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FIRE RESISTANT AIRCRAFT HYDRAULIC SYSTEM.(U)

JUL 82   E T RAYMOND, D W HULING, R L SHICK   F33615-76-C-2064

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FIRE RESISTANT AIRCRAFT HYDRAULIC SYSTEM

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JULY 1982

FINAL REPORT FOR PERIOD MAY 1976 - MAY 1980

APPROVED FOR PUBLIC RELEASE; DISTRIBUTION UNLIMITED

AERO PROPULSION LABORATORY
AIR FORCE WRIGHT AERONAUTICAL LABORATORIES
AIR FORCE SYSTEMS COMMAND
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This technical report has been reviewed and is approved for publication.

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This document reports a study to select a nonflammable hydraulic fluid for possible use in future military aircraft in which the Halocarbon Products Corporation AO-8 chlorotrifluoroethylene (CTFE) fluid was selected as the most promising fluid which meets the specified Aero Propulsion Laboratory and Aeronautical Systems Division nonflammability criteria. It also reports the results of the component compatibility tests conducted to evaluate that fluid under typical system conditions. (continued)

(BLOCK 20 continued)

The DuPont "Freon" E6.5 fluid and the Halocarbon AO-8 fluid were selected as the primary candidates from twenty fluids proposed in response to a comprehensive industry survey. Their impacts on the design, weight, and cost of hydraulic components and systems were compared, and the AO-8 fluid was selected for component compatibility tests.

Tests of static and dynamic elastomeric seals, hydraulic pumps, and a hydraulic servoactuator indicate that the AO-8 fluid has potential for use as an aircraft hydraulic fluid. However, a number of problems, stemming primarily from its high density, high cost, and effect on seal elastomers, require solution before it can be considered acceptable.

Several promising possibilities for reducing the weight penalty associated with its use have been identified. A number of specific recommendations for additional work aimed at developing affordable means to use the AO-8 fluid or a similar CTFE fluid, and for solving the elastomeric seal problem, are offered.

FOREWORD

This report was prepared by the Boeing Military Airplane Company under Project No. 31453022 for USAF Contract F33615-76-C-2064 which was conducted between May 1976 and May 1980. The work was administered under the direction of the Aero Propulsion Laboratory at the Air Force Wright Aeronautical Laboratories, Air Force Systems Command, Wright-Patterson AFB, Ohio, with Mr. W.B. Campbell (AFWAL/POOS) as Project Engineer. Mr. Ed Binns (AFWAL/POOS) provided invaluable technical guidance throughout the program. The program was also monitored by the Materials Laboratory with Mr. C.E. Snyder (AFWAL/MLBT) and Mrs. L. Gschwender from the University of Dayton Research Institute conducting some of the fluid tests, and Messrs. T.L. Graham and W.E. Berner (AFWAL/MLBT) providing seal development data and monitoring seal test results. Mr. Greg Gandee (AFWAL/POSH) provided fluid flammability data.

Boeing Military Airplane Company participants included Messrs. E.T. Raymond, Program Manager, and D.W. Huling, of the Advanced Airplane Branch in Seattle, Washington, who conducted the fluid selection study and hydraulic seal tests; Messrs. R.L. Shick, E.C. Wagner, and W.E. Willard of the Wichita Branch, who conducted the hydraulic pump and servoactuator tests; and Mr. D.C. Sullivan of the Boeing Commercial Airplane Company Materials Technology Staff who conducted some of the fluid tests and provided consultation on questions regarding fluid and other material properties.

The special support provided by the following firms which supplied the material and support noted is gratefully acknowledged:

DuPont "Freon" Products Division, and
Halocarbon Products Corporation:
  hydraulic fluid and data for the fluid selection study.
Firestone Tire & Rubber Company:
  PNF elastomer material.
Aerospace Products Division, Nichols Plant, Lord Corporation,
Parker Seal Company, and
Precision Rubber Products Company:
  hydraulic seals molded from PNF elastomer.
Sperry-Vickers Aerospace-Marine-Defense North American Group:
  hydraulic pumps and engineering support.
Borg-Warner Nuclear Valve Division:
  modified B-52 elevator servoactuator valve.
Aircraft Porous Media, Inc.:
  hydraulic filters
Pall Corporation Scientific & Laboratory Services Department:
  hydraulic filter materials compatability study and fluid particle count analyses.
Electrochemical Technology Corporation:
  hydraulic fluid electrochemical measurements.
Monsanto Research Corporation
  hydraulic fluid flammability data

In addition, the overhaul of a B-52 hydraulic servoactuator by Air Force Air Logistic Command technicians from McClellan AFB is acknowledged.

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1.0 INTRODUCTION AND SUMMARY

The Air Force incurs large dollar losses yearly because aircraft are vulnerable to fire. This vulnerability also poses a threat to human life in peace time and increases the vulnerability to enemy gunfire during combat. A number of aircraft fires, approximately ten per year over the recent past fifteen-year period (1965-79) involved the petroleum-base hydraulic fluid in general use in Air Force aircraft.

That fluid, procured to specification MIL-H-5606, has served the Air Force well for many years. It is very stable, is reasonably priced, and has the necessary properties for a good aircraft hydraulic fluid including the viscosity which allows operation over the complete temperature range of -65F to 275F as required for Type II systems per MIL-H-5440 which cover subsonic aircraft and most supersonic aircraft. However, it is very flammable when ignited. It has poor resistance to ignition from gunfire; and, many noncombat hydraulic fluid fires have also occurred. A majority of the latter have been of the hot surface type and have occurred in the wheel well areas due to fluid leaking onto hot brake assemblies.

In recognition of that experience, the Aero Propulsion Laboratory, with support from the Materials Laboratory, both of the Air Force Wright Aeronautical Laboratories, initiated a research development program for the establishment and verification of parameters for designing fire resistant aircraft hydraulic systems. That program, which is documented herein, included the following primary tasks:

a. Fluid Selection Study
b. Component Compatibility Tests
c. Design Guide Preparation

1.1 Fluid Selection

A comprehensive industry/agency survey was conducted to identify candidate materials, the responses were evaluated, and flammability tests were run to determine which nonflammable materials could be used directly as hydraulic fluids and basestocks from which a workable hydraulic fluid could be developed. The survey questionnaire was sent to 71 fluid manufacturing organizations in 37 separate companies or corporations, and 35 fluid user/research organizations in 26 separate companies, universities, or government agencies.

Fifty-two responses were received, and 20 different fluids were proposed. Sixteen of those were eliminated because of gross physical property deficiencies or extraordinary high cost. The following four were given further consideration:

Supplier	Fluid Designation
DuPont "Freon" Products Division	"Freon" E6.5 Fluorocarbon
Halocarbon Products Corporation	AO-8 Chlorotrifluoroethylene (CTFE)
Penwalt Corporation	Fluorocarbon Liquid 71
3M Commercial Chemical Division	"Fluorinert" FC-48

The DuPont "Freon" E6.5 fluid and the Halocarbon AO-8 fluid were selected as the primary candidates. The physical properties required to assess and compare the impact of each fluid upon typical aircraft hydraulic components and systems were obtained by a cooperative program of fluid testing conducted and/or sponsored by the Aero Propulsion Laboratory, the Materials Laboratory, the Boeing Company, and the fluid suppliers.

Analytical evaluations of the design and performance impact on a number of hydraulic system components anticipated by the use of each of the two candidate fluids in lieu of mineral fluid per MIL-H-5606 were made. In addition, the overall impact of each of the candidate fluids upon complete aircraft hydraulic systems, in terms of the increases in size and weight to maintain normal performance, the increase in cost, and potential changes in reliability, maintainability, and safety, were estimated.

The two fluids were found to be essentially equal in regard to their flammability properties and in many other properties evaluated for hydraulic fluid performance. Their primary differences are in their viscosity, bulk modulus, and elastomer compatibility. The density of both fluids is more than double that of MIL-H-5606 fluid. That, and their higher viscosities which require larger line sizes, combine to cause a significant increase in system weight.

Due to its lower viscosity in the normal operating temperature range, the E6.5 fluid would have a lesser impact on system weight than the AO-8 fluid. On the other hand, the low high-temperature viscosity of the E6.5 fluid might require heavier components to avoid hydrodynamic lubrication failure. Also, the lower bulk modulus of the E6.5 fluid might require larger and heavier servoactuators to obtain adequate flutter stiffness. Estimates calculated for a twin-engine cargo aircraft indicated that the weight of an E6.5 fluid hydraulic system would be more than 1,250 lb greater than the baseline MIL-H-5606 fluid system, and that an AO-8 fluid system would be over 1,700 lb heavier.

Of equal concern, was the comparative elastomer compatibility of the two fluids. The E6.5 fluid appeared to be compatible with ethylene propylene rubber which is a well proven O-ring material; but, the Materials Laboratory indicated that there was a problem finding an O-ring elastomer which was compatible with the AO-8 fluid. Although two materials appeared promising, considerable testing would be required to validate them for actual use.

Of additional concern, was the relative cost and availability of the two fluids. This was considered the deciding factor. Estimates calculated for the same twin-engine cargo aircraft indicated that the increase in life-cycle cost due to the use of the E6.5 fluid would be over $650,000 per aircraft, and that the increase due to use of the AO-8 fluid would be over $300,000 per aircraft based upon an initial price assumption of 37¢ per gallon of aircraft fuel. A subsequent estimate based upon an assumed fuel price of $1.00 per gallon indicated life-cycle cost increases of $778,800 and $488,500 respectively.

Those estimates were based on a projected production price of $210 per gallon for E6.5 fluid and $75 per gallon for AO-8 fluid. However, since that study was made, Halocarbon has revised their estimate upward to $125 per

gallon which increases the estimated increase in life-cycle cost for said twin-engine transport to $633,000. The unit price is now over twenty-five times the recent price of MIL-H-5606 fluid and approximately ten times the price of MIL-H-83282 fluid. However, it is still only sixty percent of the projected price for E6.5 fluid; and, in addition to the higher price for the E6.5 fluid, DuPont estimated that a new production facility to manufacture their fluid would require an investment cost of some $260,000,000 in 1978 dollars.

The price and investment cost for the E6.5 fluid was considered unacceptable. Therefore, the AO-8 fluid was recommended for component testing. It was further recommended that the Materials Laboratory continue its evaluation of elastomeric seal compounds in an effort to obtain a satisfactory elastomeric O-ring seal material.

1.2 Component Compatibility Tests

Following the selection of the Halocarbon AO-8 fluid for further evaluation, the following components were subjected to compatibility tests under typical aircraft hydraulic system operating conditions:

a. Static and dynamic elastomeric seals.
b. Hydraulic pumps.
c. Flight control servoactuator.

1.2.1 Elastomeric Seal Tests

The compatibility tests were primarily an evaluation of seals utilizing O-rings molded from a phosphonitrilic fluoroelastomer compound (PNF) which was developed by the Firestone Tire and Rubber Company and tested previously by the Materials Laboratory. Two groups of tests as follows were conducted in a seal test cylinder:

a. Reference tests of seals utilizing standard Buna N nitrile rubber O-rings in mineral fluid per MIL-H-5606.

b. AO-8 fluid compatibility tests of seals using "PNF" elastomer O-rings.

The Halocarbon AO-8 fluid/Firestone PNF seals performed as well as the MIL-H-5606 fluid/nitrile seals when compared by acceptable leakage levels. However, other secondary performance criteria had less than desirable results. The PNF material was found to adhere to some of the seal grooves in the metal seal glands; and, there was a brownish coating found on several bare 4340 steel parts. The rusty coloration did not generally cover all surfaces, but most parts had some portion of their surface discolored. Also, the rusty coloration had variable shades and one part had a distinct demarcation on one surface with heavy discoloration in one sector, but was unaffected in the remainder.

Based upon these results, the following recommendations were made:

1. The Aero Propulsion Laboratory and the Materials Laboratory should work with Firestone to develop a lower swelling and tougher surfaced PNF material to be incorporated in the later phases of testing.

2. The Materials Laboratory should work with the Halocarbon Products Corporation to develop a rust inhibitor additive and determine the appropriate concentration required.

1.2.2 Hydraulic Pump Tests

Two Sperry-Vickers Model PV3-075-15 3,000-psi variable-delivery axial-piston pumps were run under various pump qualification test conditions per specification MIL-P-19692C to determine compatibility with AO-8 fluid from the standpoints of performance and life. The results of a fifty-hour test on the first pump, which had been previously broken in on MIL-H-5606 fluid, indicated that no major redesign was required to obtain full delivery flow with acceptable power input levels with the AO-8 fluid, but that higher inlet pressures than used with MIL-H-5606 fluid are required to obtain acceptable performance. Transient discharge pressure peaks and flow response appeared to be acceptable; and, pressure pulsations were well within specified limits.

Post-test inspections showed the following adverse effects of the AO-8 fluid on the pump's internal parts:

a. The carbon shaft-seal element showed evidence of fluid attack and emitted a tar-like substance.

b. Etching and pitting of rolling-element bearings indicated insufficient lubrication.

c. A brownish discoloration or coating had formed on internal bronze and steel parts. The steel parts are primarily non-corrosion-resistant alloys; and, the parts may have rusted after having been dewetted of all protective fluid films and attacked by moist ambient air.

For future tests, it was recommended that the carbon shaft seal element be replaced with a bronze part, that a lubricity additive be added to the AO-8 fluid, and that, in future teardowns, extreme care be taken to avoid prolonged exposure to the atmosphere. It was believed that some of the etching, pitting, and discoloration may have occurred due to moisture attack following removal of the fluid film.

The lubricity additive, Molyvan A, was added to the fluid, the foregoing recommendations were applied to the second pump, and it was also given an initial fifty-hour test. The subsequent teardown inspection showed no adverse effects; and, testing was continued. Performance and endurance tests similar to those specified in MIL-P-19692C were conducted except that, in the endurance tests, the pump was run initially at 50 percent of rated speed and was gradually brought up to full rated speed in incremental steps.

A failure occurred after a total accumulated running time of 713 hours 49 minutes (650 hours and 13 minutes of the specified 750-hour endurance test) with the pump running at its rated speed of 7,000 rpm. The subsequent teardown inspection revealed that the pump was not repairable. The primary damage was on the bearing pads of the cylinder block and in the valve plate which suffered two cracks indicating that its surface temeprature had apparently reached 1300F.

The supplier indicated that the pump could be made to operate satisfactorily in AO-8 fluid by derating its rated speed or by rebalancing the shoes and cylinder block. The cylinder block redesign would probably involve increasing the width of the bearing pads which would, in turn, increase the cylinder block and housing size. The most cost-effective solution would be to derate the pump speed. However, this means that, for a given pump delivery flow, a larger displacement unit would be required.

The first pump, which had suffered apparent fluid attack during the first fifty-hour test, was rebuilt with a new cylinder block and piston/shoe subassemblies and was used as the pressure source for the servoactuator tests reported herein. Following the termination of those tests, that pump was used for high-temperature pump performance tests and cold-start testing all of which were satisfactorily completed.

1.2.3 Servoactuator Tests

A B-52G/H elevator servoactuator was also tested with AO-8 fluid to determine its effect upon actuator performance and life. The actuator was first modified with a new primary control valve with the metering slot widths increased approximately fifty percent, as determined by a computer simulation, to obtain gain values equal to those obtained when operating with MIL-H-5606 fluid. The modified valve did maintain the required flow gain; and, performance, including dynamic response, was generally acceptable.

The unit was endurance cycled with various input command signals for a total of approximately 2,500,000 stroke cycles. All metal parts in the valves and actuating cyclinder survived without damage except for some internal surface discoloration which did not appear to be detrimental. However, repeated test delays, due to fluid leakage from seal-induced housing cap failures attributed to excessive swell of the elastomeric O-rings, were experienced; and, the test was terminated midway through the planned schedule.

1.3 Design Guide

A design guide (Reference 1) was also prepared to document the major physical properties of the Halocarbon Products AO-8 (CTFE) fluid, and the special considerations which must be observed in the design of hydraulic systems and components intended for use with that fluid. It includes viscosity curves for two lower-viscosity CTFE fluids which may be considered as a means to reduce system weight penalties. Properties of the standard petroleum-based hydraulic fluid per specification MIL-H-5606 are also included for comparison; and, the special design considerations covered therein are primarily those which differ from those used for designing systems and components for use with MIL-H-5606 fluid. The design guide should be referred to for the current properties of the AO-8 fluid. The physical and chemical properties shown in the following pages herein do not always reflect the final fluid formulation.

1. E. T. Raymond, Design Guide for Aircraft Hydraulic Systems and Components for Use With Chlorotrifluoroethylene Nonflammable Hydraulic Fluids, AFWAL-TR-80-2111, Boeing Military Airplane Co., Seattle, WA, March 1982.

1.4 Overview Summation

The objectives of the program were met. Candidate nonflammable fluids were selected, physical properties were determined and materials compatibility tests run. The impacts upon the design of components and systems for use with the two leading candidate fluids were determined and compared. One fluid was selected for further evaluation, component compatibility tests were run, and a design guide prepared.

The selected fluid, Halocarbon Products' AO-8 CTFE polymer, meets the basic flammability and physical property requirements. Its use could significantly reduce or completely eliminate hydraulic fluid fires depending upon how extensively it is employed. However, it also has a number of serious shortcomings, including the following, which are of major concern:

a. High density.

The density of this fluid, which is more than double that of the hydrocarbon fluids, requires larger component orifices and valve slots, the derating of hydraulic pump speeds, increased reservoir pressures, and, combined with its higher viscosity, larger line sizes than would normally be required. These factors, and the increased weight of the fluid itself, all add up to a significant increase in system weight if the fluid is used throughout a system of the conventional type.

b. High cost.

The high cost of the fluid, the increase in costs for component development, the increase in investment costs for new hydraulic benches and ground carts, and the increased use of aircraft fuel due to the increase in system weight, all add up to a significant increase in aircraft life-cycle cost if the fluid is used throughout a system of the conventional type.

c. Marginal elastomer compatibility.

Unless this problem is resolved, non O-ring type seals may be required to avoid chronic seal problems. This will increase costs further; or, if O-rings are used, system reliability may drop and maintenance may rise.

Of additional concern is the reaction of the fluid with bronze alloys. The long-term effects are as yet unknown, but the preclusion of bronze would be very serious indeed. It is generally considered the best material for critical bearing applications in essential hydraulic components.

In spite of these concerns, it is concluded that the Halocarbon AO-8 fluid is the best potential hydraulic fluid material available to meet the Air Force nonflammability requirements, but that additional development and evaluation is required before it can be recommended for aircraft use. Several promising possibilites for reducing the weight penalty have been identified; and, a number of specific recommendations for additional work aimed at developing affordable means to use the AO-8 fluid, or a similar CTFE fluid, are offered. It is recommended that the development of the two-fluid fireproof brake hydraulic system concept be continued. See the specific recommendations in Section 5.0.

2.0 FLUID SELECTION STUDY

Previous Air Force efforts to develop a nonflammable hydraulic system have been unsuccessful due to the constraints heretofore imposed by the requirement that a new fluid must be compatible with both MIL-H-5606 fluid and present-day hydraulic systems and components. When those constraints were lifted, it allowed the consideration of materials other than petroleum-base fluids or synthetic hydrocarbons. In addition, the nonflammability requirements imposed by the Aero Propulsion Laboratory and Aeronautical System Division personnel were so stringent that only a very few materials could qualify.

2.1 FLUID PROPERTY REQUIREMENTS

Two types of criteria were established: minimum requirements necessary for initial consideration, and target values for various individual properties.

2.1.1 Minimum Requirements for Candidate Fluids

In order to qualify for consideration as a nonflammable hydraulic fluid with capability for operation at system temperatures from -65 to 300F in future Air Force aircraft, candidate fluids were required to meet the following minimum requirements:

Property	Test Method	Value
Autogenous Ignition Temperature	ASTM D-2155 (Modified to include injection pressure to 1000 psig)	>1300F
Hot Manifold Ignition Temperature	Modified Federal Test Standard 791B - Method 6053	>1700F
Heat of Combustion	ASTM D-240 (Bomb Method)	<5000 BTU/LbM
Atomized Fluid Flammability Test	Aero Propulsion Laboratory Procedure as follows:	self extinguishing flame

It was required that the fluid be sprayed through a 70-degree spray cone, 2.25-gallons-per-hour oil burner nozzle (used in home oil burners) that has been drilled out to .016" diameter. The nozzle pressure and temperature were held at 300 psig and 65+10F, respectively in a system capable of delivering fluid at these conditions for at least three minutes after the ignition has been applied. The ignition sources were (1) 6-joule 20K volt spark, (2) 6-inch high pre-mixed, stoichiometric propane-air flame emanating from 3/4" I.D. burner, and (3) incendiary gunfire simulator (AFAPL-TR-73-50, Incendiary Gunfire Simulation Techniques for Fuel Tank Explosion Protection Testing). These ignition sources were held two feet from the nozzle and intersecting its spray centerline. The success criteria for the spark ignition source and the simulated incendiary gunfire was that the fluid may flash, however, the flame must be self-extinguishing. The success criteria for the propane-air flame ignition source was that the fluids flame front does not propagate back to the nozzle and that the flame must be self-extinguishing when the ignition source is removed.

2.1.2 Target Property Values

In addition to the foregoing minimum requirements, target values were established for a number of other fluid propertiees. The initial target values were as follows. However, as it became apparent that only relatively high density fluids could meet the flammability requirements, some of the target values were subsequently revised.

a. Pour Point

To be usable at the specified minimum system operating temperature of -65F, the fluid pour point should be below -75F.

b. Viscosity

To be usable throughout the specified system operating temperature range of -65F to 300F, a fluid viscosity no greater than 2500 centistokes at -65F and no less than 2 centistokes at 300F was desired. Exceptions would be considered provided the fluid was considered usable throughout the specified temperature range without entailing undue penalties in system design and/or operational performance. The target values were later revised to 5500 and 1.0 centipoise respectively.

c. Thermal Stability

Stability at 300F in the presence of typical system metals was required. Data indicating high temperature instability, such as evidence of undue change in a fluid's viscosity or increase in acid number, or in weight loss or other evidence of corrosive attack on typical system materials could be cause for rejection. It was desired that each viable candidate be stable in the presence of all the following materials, but some exceptions would be allowed if material substituations could be made without undue penalty:

bare carbon steels, stainless steels, bearing steels, aluminum, beryllium copper, bronze, and titanium alloys, chrome, cadmium, electroless nickel, and silver platings.

d. Toxicity

Inasmuch as aircraft hydraulic fluids must be safely handled by maintenance personnel without protective equipment, it was required that no health hazard or cumulative toxic effect result from skin contact with any fluid proposed or from the breathing of its vapors under normal handling conditions.

2.2 CANDIDATE FLUIDS

A comprehensive industry/agency survey was conducted to identify candidate materials. The responses were evaluated; and, of the twenty different fluids proposed, four were selected for further consideration. Of those, two were selected as the primary candidates.

2.2.1 Industry/Agency Survey

The survey questionnaire, included in Appendix A, was sent to the 106 addressees noted therein. They included 71 fluid manufacturing organizations in 37 separate companies or corporations, and 35 fluid user/research organizations in 26 separate companies, unviersities, and government agencies. Fifty-two responses were received and the twenty different fluids tabulated in Table 1 were proposed.

2.2.2 Selection of Four Initial Fluid Candidates

Chevron's Fluid A was deleted because of its toxicity, General Electric's "Versilube" F-50 and SF-1148 because of their inability to meet the flammability requirements and the Dow Chemical's "Alkazene" 42 which is not likely to meet the flammability requirements. "Alkazene" 42 is a dibromoethyl-benzene whose molecule may be characterized as having many hydrogen atoms, which are known to increase ignitability.

The following fluids were deleted because of their high low-temperature viscosity: Du Pont Petroleum Chemical Division's "Krytox" 143AZ and 143CZ, Pennwalt Corporation's Fluorocarbon Liquid 91, 3M Commercial Chemical Division's "Fluorinert" FC-70, and PCR's monotriazines 4m4p, 3m3p, and 3m2p. The least viscous of the "Krytox" fluids is 143AZ with 150,000 centipoises (75,000 cs) at -65F. Pennwalt's Liquid 91 has a freezing point of -44F, and 3M's FC-70 has a -13F pour point. The three monotriazines listed from PCR have -65F viscosities in the 25,000 to 40,000 centipoise (12,000 to 20,000 cs) range. These fluids with very high viscosities within the operating temperature range, would unnecessarily compromise system design since chemically similar fluids with less than 10,000 centipoise viscosities were proposed. However, the possibility exists that these fluids could be used in blends with thinner base fluids.

Several fluids were deleted because of their extraordinary high cost. Montedison's Fomblin Z-15 was priced at $1300 per gallon in large quantity production (10,000 gallons per year), and the following statement was made by Montedison about Fomblin Z-04: "We have given no cost indication for the Fomblin Z-04 fraction becasue we think it would at present be out of a conceivable range, . . . ". Fomblin Z-15, although its low temperature viscosity is acceptable, has a flat viscosity curve characteristic that indicates an excessive distribution system inefficiency (line losses) in the normal operating temperature range. The Fomblin Z-04 is much better in this respect with a viscosity-temperature characteristic nearly equal to MIL-H-5606. A third fluid, proposed by Bray Oil Co., Brayco 814Z, had a price quote of $2000 per gallon for large production quantities. PCR, in addition to their three fluids previously mentioned with excessive viscosity, presented two fluids with good viscosity characteristics. These fluids 4mlp and 5mlp, although promising, are expected to greatly exceed $2500 per gallon in high production quantities.

The remaining four fluids, Du Pont's "Freon" E6.5, Halocarbon's AO-8, Pennwalt's Liquid 71 and 3M's "Fluorinert" FC-48 were selected for further consideration. However, the "Fluorinert" FC-48 and Liquid 71 would require significant development to approach some of the established acceptance criteria. For instance, both fluids exhibit very high vapor pressures and

TABLE 1 THE TWENTY FLUIDS PROPOSED IN RESPONSE TO THE INDUSTRY SURVEY

Bray Oil Company

"Brayco" 814z - Perfluoropolyether

Chevron International Oil

"Oronite" Fluid A - Chlorinated Hydrocarbon

Dow Chemical U.S.A., Texas Division

"Alkazene" 42 - Dibromoethylbenzene

Du Pont, "Freon" Product Division

"Freon" E6.5 - Fluorocarbon

Du Pont, Organic Chemical Department

"Krytox" 143AZ - Perfluoroalkylpolyether

"Krytox" 143CZ - Perfluoroalkylpolyether

General Electric, Silicone Products Department

SF-1148 - Methalkyl Polysiloxane

"Versilube" F-50 - Methyl Chlorophenyl Polysiloxane

Halocarbon Products Corporation

AO-8 - Chlorofluorocarbon

3M, Commercial Chemical Division

"Fluorinert" FC-48 - Fluorinated Hydrocarbon

"Fluorinert" FC-70 - Fluorinated Hydrocarbon

Montedison S. p. A.

"Fomblin" Z-04 - Perfluoropolyether

"Fomblin" Z-15 - Perfluoropolyether

Pennwalt Corporation

Fluorocarbon Liquid 71

Fluorocarbon Liquid 91

PCR, Inc., Contract Research Division

4m1p Monotriazine - monotriazine

5m1p Monotriazine - monotriazine

4m4p Monotriazine - monotriazine

3m3p Monotriazine - monotriazine

3m2p Monotriazine - monotriazine

Liquid 71 has a very low viscosity. The "Fluorinert" FC-48 may have an undesirably low viscosity at higher temperatures as indicated by data extrapolation. The physical properties information available on these four fluids at that time is presented in Table 2.

2.2.3 Section Of Two Finalist Fluid Candidates

The Du Pont "Freon" E6.5 and Halocarbon AO-8 fluids were selected as the primary nonflammable fluid candidates. Pennwalt Liquid 71 and 3M "Fluorinert" FC-48 were held in abeyance for substitution if either or both of the primary fluid candidates were found to contain an unacceptable property which couldn't be surmounted.

The Du Pont "Freon" E6.5 fluorocarbon fluid was recommended because it has most of the physical properties required of a hydraulic fluid. The molecular structure of the fluid's 1200-average-atomic-weight molecule contains only one hydrogen atom. This, coupled with a "non-additives" formula, indicates a high degree of fire resistance. E6.5's low-temperature viscosity, estimated to be 5,700 centipoises, exceeds the 5,500 centipoise target, but would not excessively compromise most overall system designs. The pour point was stated to be -90F, well below the "lower than -75F" target acceptance criteria. The bulk modulus is lower than the target acceptance criteria, however, other properties information tended to indicate acceptability.

Halocarbon's AO-8 chlorotrifluoroethylene (CTFE) fluid was also recommended because it was expected to meet the nonflammability criteria, and it appeared to have acceptable hydraulic fluid characteristics. The low-temperature viscosity, Shell four-ball, and vapor pressure data, although exceeding the target values, were within the state-of-the-art design capabilities. Elastomer compatibility was expected to be the greatest deficiency of this fluid.

2.3 COMPARISON OF THE TWO FINALIST FLUID CANDIDATES

In order to select one of the two candidates for evaluation tests with typical aircraft components, additional physical property data was gathered and analyses were conducted to estimate their relative impact on hydraulic component and system performance and design, and on system cost, reliability, maintainability, and safety.

2.3.1 Fluid Property Data

The properties used to assess and compare the impact of the two candidate fluids upon component and system performance and design are summarized in Table 3. Values for a number of those properties (designated by the letter A) were provided by the Aero Propulsion Laboratory or the Materials Laboratory, others (designated by the letter B) were determined by the Boeing Materials Technology Laboratory, and others (designated by the letter C) were provided by the fluid suppliers. Comparable properties of the standard petroleum-based hydraulic fluid per MIL-H-5606 and the synthetic hydrocarbon hydraulic fluid per MIL-H-83282 are also shown in Table 3.

Curves of density, viscosity, vapor pressure, specific heat, thermal conductivity, and electrical conductivity for the two candidate fluids and for

TABLE 2 AVAILABLE PROPERTIES OF THE FOUR INITIAL FLUID CANDIDATES

Parameter	Target Value	Du Pont "Freon" E6.5	Halocarbon AO-8[a]	3M "Fluorinert" FC-48	Pennwalt Fluorocarbon Liquid 71
Chemical Type		Fluorocarbon	Chlorofluoro-carbon	Fluorinated Hydrocarbon	Fluorocarbon
Flammability	AIT >1300F Hot Manifold Ign. Temp >1700F Heat of Comb. <5000 BTU/lb.	One hydrogen molecule No Flash Pt.	Non-hydrogen molecule Pass Heat of Combustion	Non-hydrogen molecule No Flash Pt.	Non-hydrogen molecule No Flash Pt.
Density @ 77F		1.82 gm/cc	1.84 gm/cc	1.94 gm/cc	1.77 gm/cc
Viscosity @ -65F	<5500 cp	5700 cp	2460 cp[b]	25,000 cp[b]	10 cp
@ 240F	>3.3 cp	2.2 cp	3.4 cp	1.0 cp[b]	0.7 cp
Pour Point	< - 75F	- 90F	< - 95F	- 80F	- 175F
Vapor Pres @ 300F	< 30 mm of Hg.	25 mm of Hg.[b]	370 mm of Hg.[b]	300 mm of Hg.[b]	5000 mm of Hg.[b]
Thermal Stability	> 300F	to 500F	to 400F	Excellent to 300F	to 750F
Toxicity	≤ MIL-H-5606	Pass		Pass	Vapors and orally non-toxic to mice
Price/Gallon In High Prod. (1976 Estimate)		\$75[b]	\$120[b]	\$200	\$390

[a] Preliminary blend of Halocarbon AO-8 fluid [b] Data later revised

TABLE 3 SHEET 1 PROPERTIES OF THE TWO CANDIDATE FLUIDS AND MIL-H-5606 AND MIL-H-83282 FLUIDS

	TARGET VALUE	DuPONT "FREON" E6.5	HALOCARBON AO-8	MIL-H-5606	MIL-H-83282
CHEMICAL GROUP		FLUOROCARBON	CLORO-FLUOROCARBON	PETROLEUM	SYNTHETIC HYDROCARBON
AUTO-IGNITION TEMPERATURE (A)	>1300 F	1240 F (Transient small blue flame)	1170 F (Transient small blue flame)	450 F	700 F
HOT MANIFOLD IGNITION TEMP (A)	>1700 F	PASS	PASS	FAIL	FAIL
HEAT OF COMBUSTION (BTU/#) (A)	<5000	PASS	PASS	FAIL	FAIL
ATOMIZED FLUID FLAMMABILITY (A)	Any flame must self extinguish	PASS	PASS	FAIL	FAIL
DENSITY @ 77F (gm/cc) (A)		1.82	1.86	.84	.83
COEFFICIENT OF (B) THERMAL VOL. EXP. $IN^3/IN^3/F$		4.8×10^{-4}	5.0×10^{-4}	4.0×10^{-4}	4.6×10^{-4}
VISCOSITY (cp) (A)					
@ -65F	<5500	5700	4870	2050	10,250
@ 240F		2.2	3.1	2.9	2.1
@ 300F	>1.0	1.5	2.1	2.1	1.5
POUR POINT (C)	<-75F	-90F	<-95F	-85F	<-75F
LOW TEMP STABILITY	No clouding or solids to -65F	PASS (A)	PASS (C)	PASS	PASS

TABLE 3 SHEET 2 PROPERTIES OF THE TWO CANDIDATE FLUIDS AND MIL-H-5606 AND MIL-H-83282 FLUIDS

	TARGET VALUE	DuPONT "FREON" E6.5	HALOCARBON AO-8	MIL-H-5606	MIL-H-83282
THERMAL STABILITY (AMINCO BOMB) (A)	>300F	PASS	PASS		PASS
OXIDATIVE STABILITY (AMINCO BOMB) (A)	>300F	PASS	PASS	PASS	
HYDROLYTIC STABILITY (AMINCO BOMB) (A)	>300F	PASS	PASS	PASS	
OXIDATION STABILITY (FED. STD.) (A)	MIL-H-5606 Reqm't.	PASS	PASS	PASS	PASS
LUBRICITY - SHELL 4-BALL @ 1200 RPM, 167F & 1 HR. (A)					
1 Kg (mm)		0.22	0.21		
10 Kg (mm)	<0.5	0.34	0.34	.50	.25 (600 RPM)
40 Kg (mm)	<1	0.60	0.87	1.00	.48 (600 RPM)
BULK MODULUS, ADIABATIC TANGENT @77F & 3000 PSI (B)	≥ 200,000 PSI	157,300 PSI	240,700 PSI	273,300 PSI	274,200 PSI
VAPOR PRESSURE (mm Hg.) (A)					
@ 210F	<5	3	6	9.5	3.5
@ 240F		6	15	19	5.0
@ 300F	<30	22.5	71	56	8.5
FOAMING (A)	MIL-H-5606 Reqm't.	PASS	PASS	PASS	PASS
VALVE STICTION (B)					
MAX VALVE FORCE	< 5 LBS.	0.5 LBS.	1.0 LBS.	<0.1 LBS.	<0.1 LBS.
TAN (before/after)	≤.2		.05/.05	.02/.05	.05/.01
KIN. VIS. @ 100°F *		5.2/6.0 CS.	7.3/70.0 CS	14.4/20.6 CS	15.7/15.8 CS
Δ%	±5%	+15%	+859%	+43%	+.6%

* before/after

TABLE 3 SHEET 3 PROPERTIES OF THE TWO CANDIDATE FLUIDS AND MIL-H-5606 AND MIL-H-83282 FLUIDS

	TARGET VALUE	DuPONT "FREON" E6.5	HALOCARBON AO-8	MIL-H-5606	MIL-H-83282
ELASTOMER COMPATIBILITY (A)		"VITON" FLUORINATED PHOSPHONITRILIC ETHYLENE PROPYLENE	CHLOROPOLY-ETHYLENE FLUORINATED PHOSPHONITRILIC	L STOCK BUNA-N	L STOCK BUNA-N
ELECT. INSULATION COMPATIBILITY (C)		TFE 500F FEP to 400F POLYARYLENE >100C FLUOROPOLYMER >100C	TFE FEP POLYIMIDE PVF_2 POLYARYLENE	NORMAL MATERIALS	NORMAL MATERIALS
COPPER STRIP CORROSION TEST (A)	$\leq$ 2e, MIL-H-5606		FAILS, 3b	PASS	PASS
ELEC. CONDUCTIVITY (mho/cm)	$<5X10^{-8}$ $>3X10^{-7}$ VALVE EROSION BAND	$5X10^{-16}$ (C)	$<<1.5X10^{-8}$ (B)	$9X10^{-10}$	$1.2X10^{-10}$
SPECIFIC HEAT (A) $(\frac{BTU}{\#^{o}F})$ @ 100F		.245	.234	.47	.50
THERMAL COND. (A) $(\frac{BTU}{HR.FT.^{o}F})$ @ 100F		.045	.043	.078	.097
SONIC SHEAR (A)	PASS	NOT TESTED BUT EXPECTED TO PASS	PASS	PASS	PASS
TOXICITY (C)	$\leq$MIL-H-5606	PASS	PASS	PASS	PASS
PRICE @ 10^6 GAL/YR (1976 DOLLARS) (C)		\$200/GAL.	\$55-75/GAL.	\$3/GAL.	\$7/GAL.

CANDIDATE FLUID DATA SOURCE (A) AFWAL (B) BOEING (C) SUPPLIER

MIL-H-5606 fluid are shown in Figures 1 through 6. The data for some of the MIL-H-5606 fluid properties were taken from SAE Aerospace Information Report AIR 1362 (Reference 2). Again, it should be noted that many of the Halocarbon AO-8 fluid property values shown herein are for an early formulation, and that the design guide (Reference 1) should be consulted for the latest values available during this program.

2.3.2 Fluid Tests

Early in the Phase I effort, a cooperative program of fluid testing was agreed upon between the Aero Propulsion Laboratory, the Materials Laboratory, and Boeing so as to make the best use of each laboratory's equipment and capabilities. Since the primary acceptance criteria deals with the flammability characteristics, the Aero Propulsion Laboratory first set out to verify that the two candidate fluids met the designated limits.

When these results were found to be satisfactory, the Materials Laboratory started examining the known deficiencies such as the incompatibility of standard elastomeric seal materials with both candidate fluids, the high vapor pressure of the AO-8 fluid, and the low viscosity of the E6.5 fluid at high temperatures. At the Materials Laboratory request, the Halocarbon Company was able to provide several new fluids with successively reduced vapor pressure and increased lubricity. However, it was found that all known viscosity improvers are insoluble when mixed with the E6.5 fluid which prevented any improvement in its high-temperature viscosity.

In addition to the tests run to determine the fluid properties normally required for aircraft hydraulic fluids, the following tests were run to provide additional bases for evaluating and comparing the two candidate fluids and to determine if there were any unacceptable deficiencies.

2.3.2.1 Flammability Tests

The Aero Propulsion Laboratory contracted with the Monsanto Research Laboratory to develop apparatus for the measurement of ignitability characteristics of fluids at high temperatures (up to 1700F), and to use that and other apparatus for the determination of ignitability and flame propagation properties and heats of combustion of a number of aircraft fluids. The results were documented in Reference 3.

The autoignition temperatures of the two candidate fluids and the two reference fluids are shown in Table 3, herein, along with the results of the hot manifold ignition, heat of combustion, and atomized fluid flammability tests. The autoignition temperatures of both candidate fluids closely approached the 1300F requirement and were well above the AITs of the reference fluids. Both candidate fluids passed the other three flammability tests whereas both reference fluids failed.

2. SAE AIR 1362, Aerospace Information Report, Physical Properties of Hydraulic Fluids, Society of Automotive Engineers, Inc., Warrendale, PA, May 1975.

3. Leo Parts, Assessment of the Flammability of Aircraft Hydraulic Fluids, AFAPL-TR-79-2055, Monsanto Research Corporation, Dayton, Ohio, July 1979.

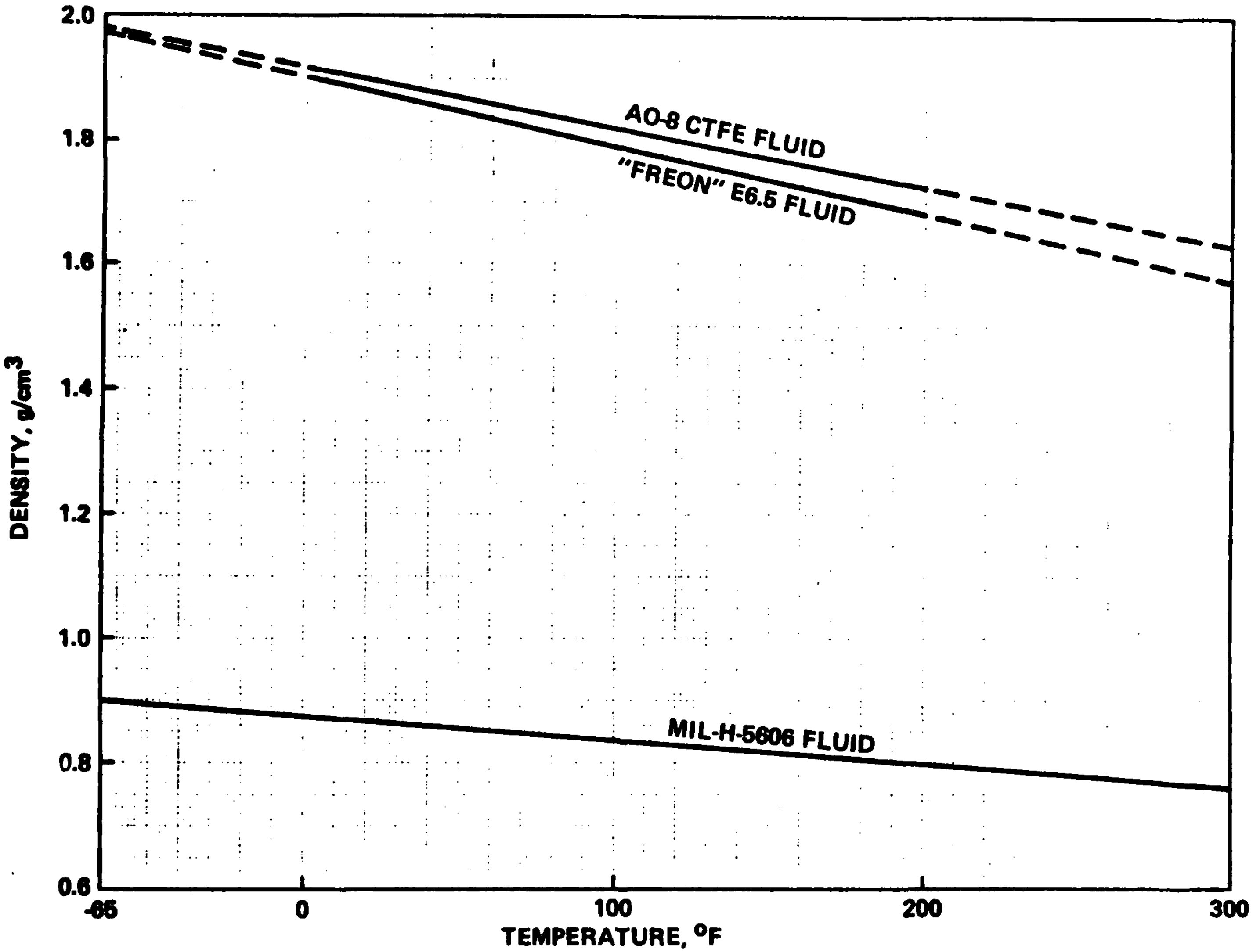

Figure 1. Density of the two candidate fluids and MIL-H-5606 fluid

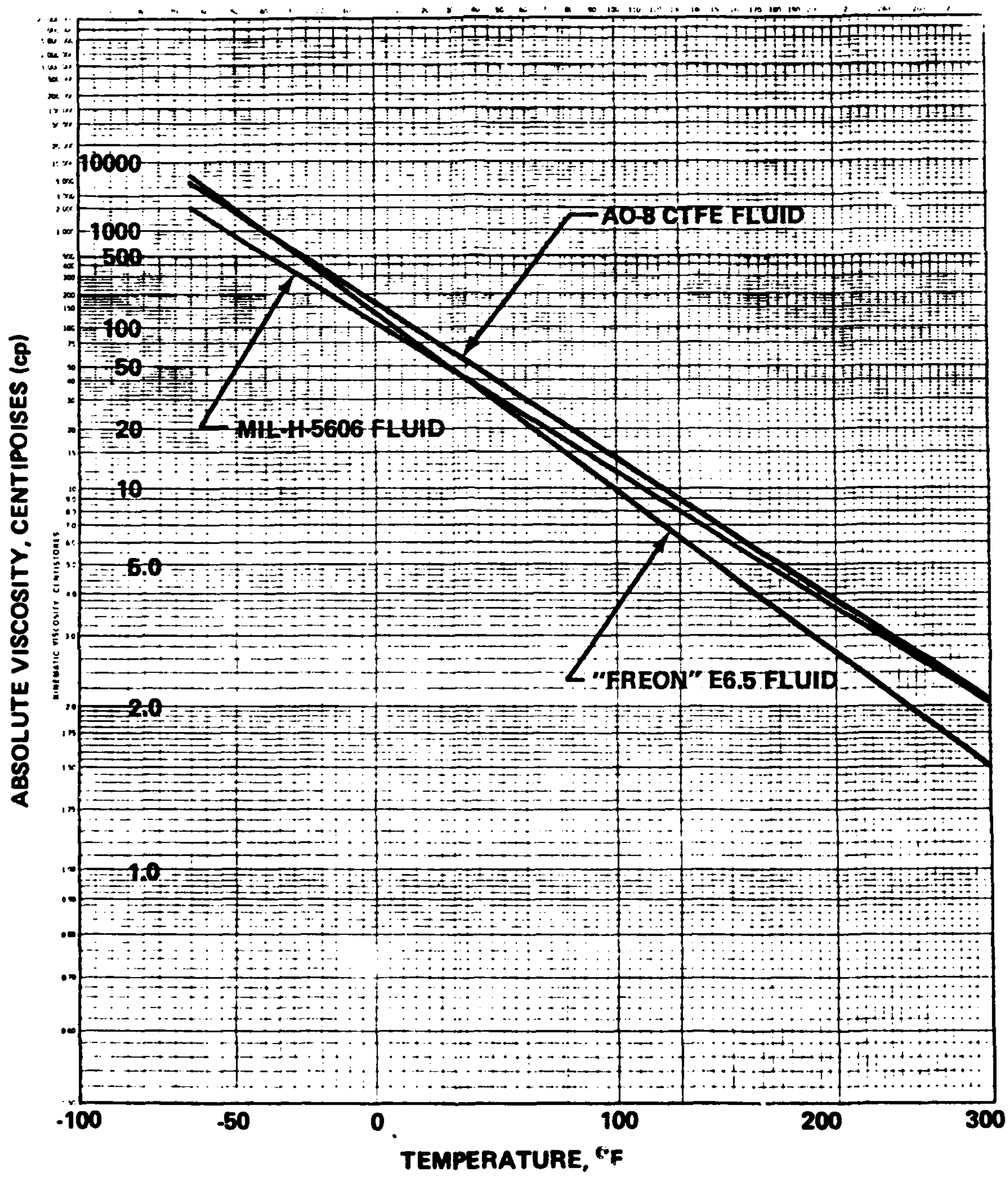

Figure 2. Viscosity of the two candidate fluids and MIL-H-5606 fluid

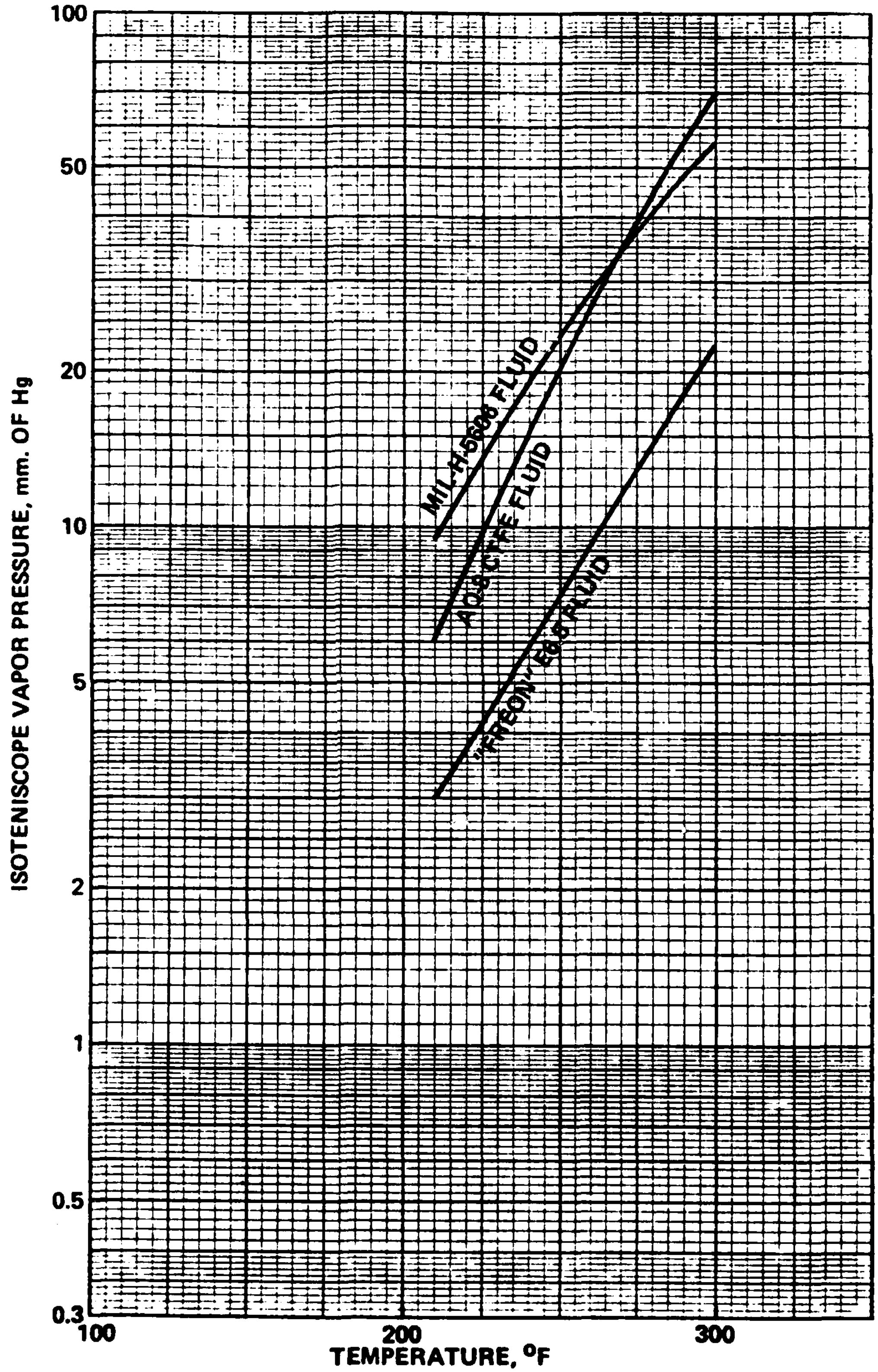

Figure 3. Vapor pressure of the two candidate fluids and MIL-H-5606 fluid

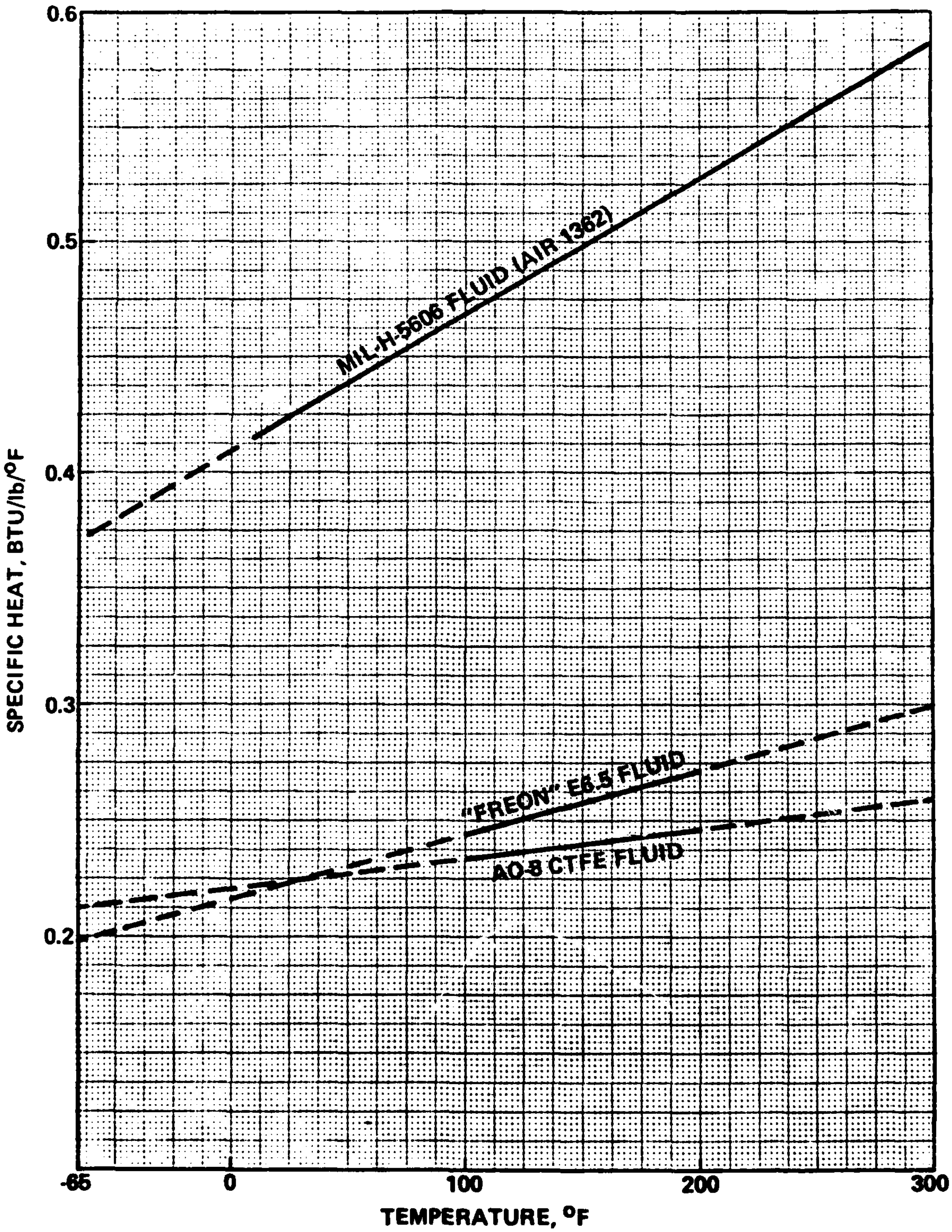

Figure 4. Specific heat of the two candidate fluids and MIL-H-5606 fluid

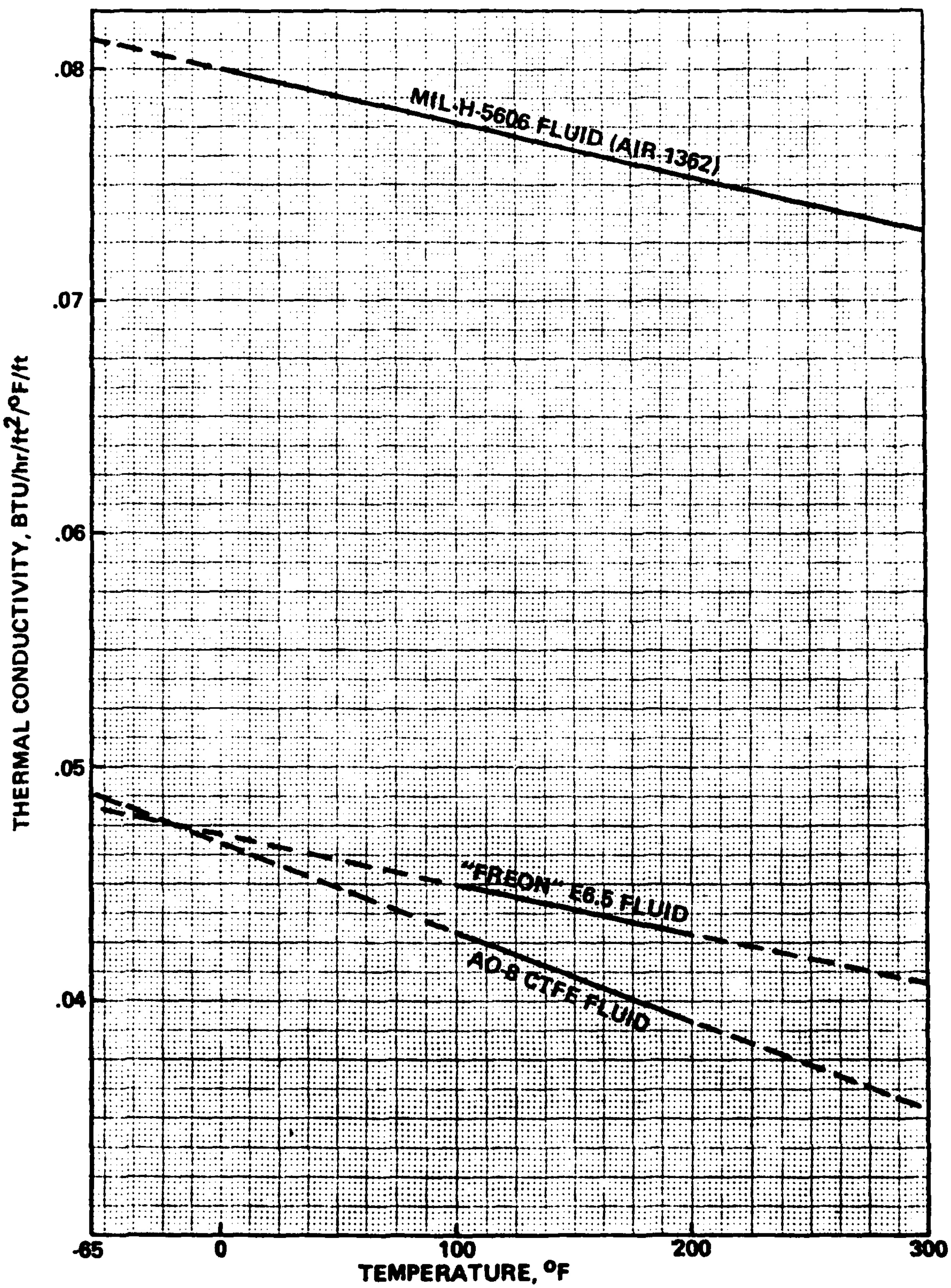

Figure 5. Thermal conductivity of the two candidate fluids and MIL-H-5606 fluid

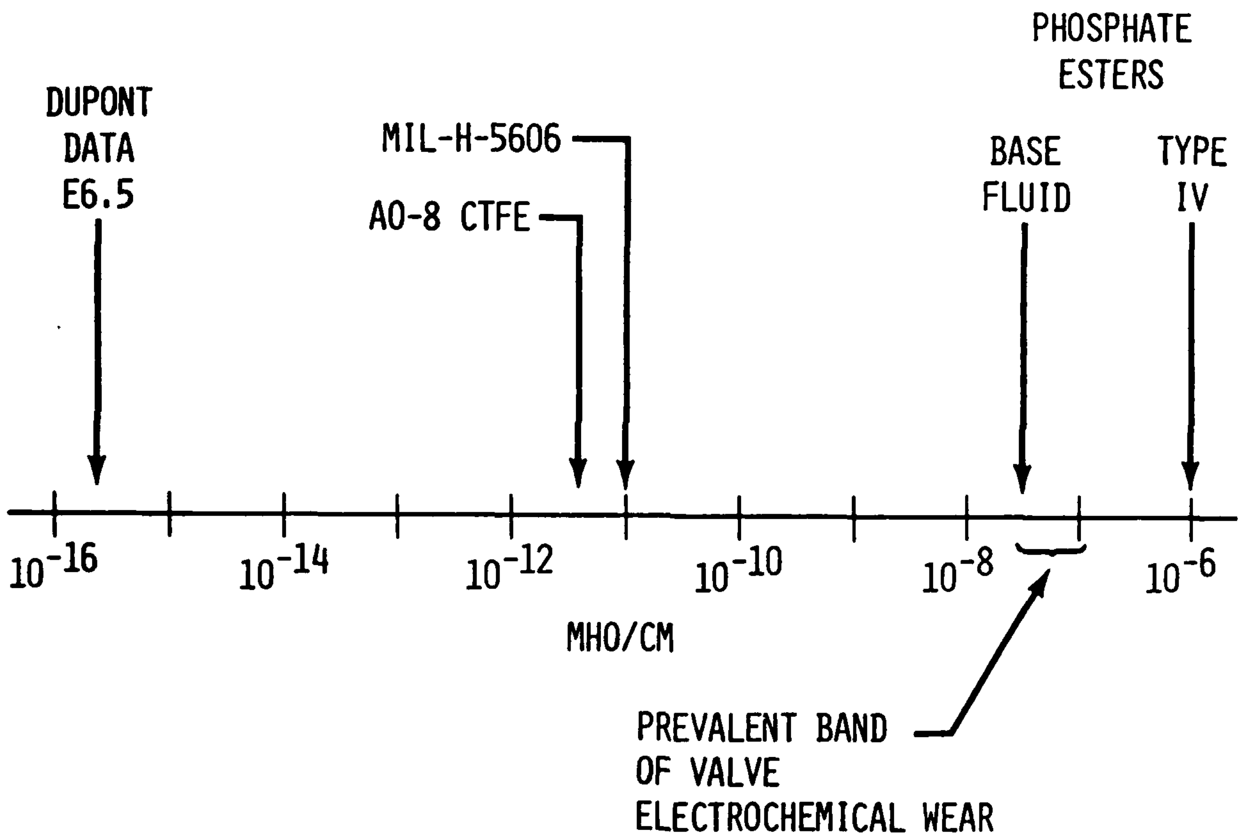

Figure 6. Electrical conductivity of the two candidate fluids and standard hydraulic fluids

2.3.2.2 Bulk Modulus Tests

Adiabatic tangent bulk modulus was measured at Boeing by the speed of sound method with the NUS Corp. Model 6000 Bulk Modulus Test Unit to determine a 3000-psi room-temperature value for both candidate fluids. Values were also determined for the four reference fluids noted in Table 4 for use in comparisons with available values determined by other laboratories and other methods.

TABLE 4 COMPARATIVE BULK MODULUS VALUES
(Adiabatic tangent values at 3,000 psi and 77F)

Candidate Fluid	Bulk Modulus - PSI
DuPont "Freon" E6.5	157,300
Halocarbon AO-8 CTFE	240,700
Reference Fluid	
Petroleum Fluid per MIL-H-5606	273,300
Synthetic Hydrocarbon per MIL-H-83282	274,200
Deep Dewaxed Hydrocarbon per MIL-H-27601	278,700
Silicate Ester Fluid M2V	258,500

In addition, following the selection of the Halocarbon AO-8 fluid as the single fluid for further testing, bulk modulus measurements were made at 72.5F, 150F, and 250F and at pressures of zero, 1500, 3000, and 4500 psig. These data are presented in the design guide, Reference 1, along with comparative data for MIL-H-5606 fluid.

2.3.2.3 Valve Stiction Tests

Valve stiction testing was also completed at Boeing on the candidate and reference fluids. The Boeing valve stiction test, which was developed on the SST program, is intended to determine the ability of a fluid to deteriorate without forming substances such as varnish which could seize a spool valve or other small-clearance sliding surfaces. The test setup is pictorially described in Figure 7. The fluid was cycled through a temperature range of 100F to 300F per Figure 8 to simulate a fluid thermal cycle in a typical aircraft system. The valve specimen was checked for stiction each day for the duration of the test; and at 100-hour intervals, the valve was removed from the sealed system and exposed to air for two hours to simulate component and system maintenance. The test was complete at 800 hours (200 thermal cycles) and a check of fluid properties made. The results are summarized in Table 5.

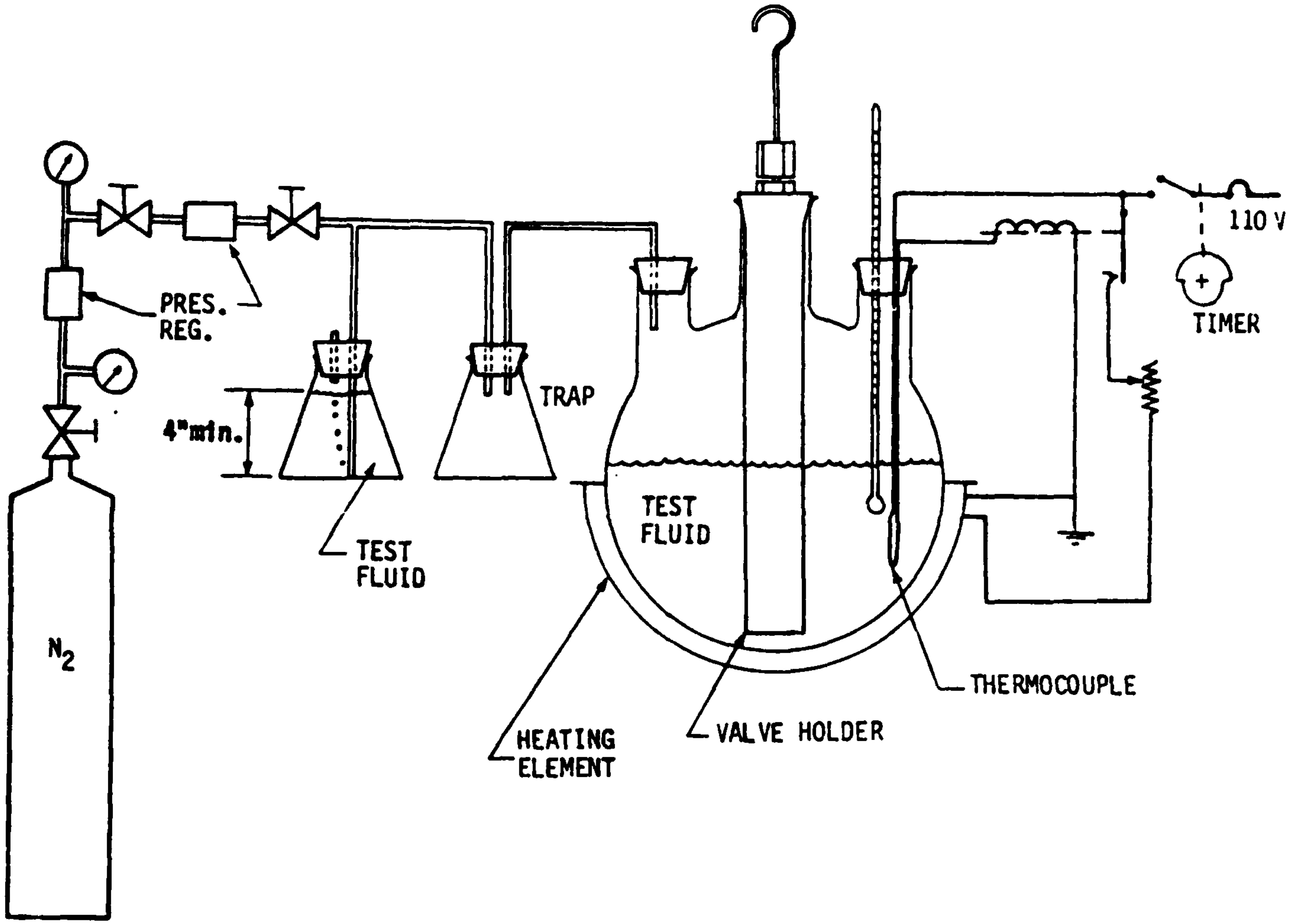

Figure 7. Valve stiction test setup schematic

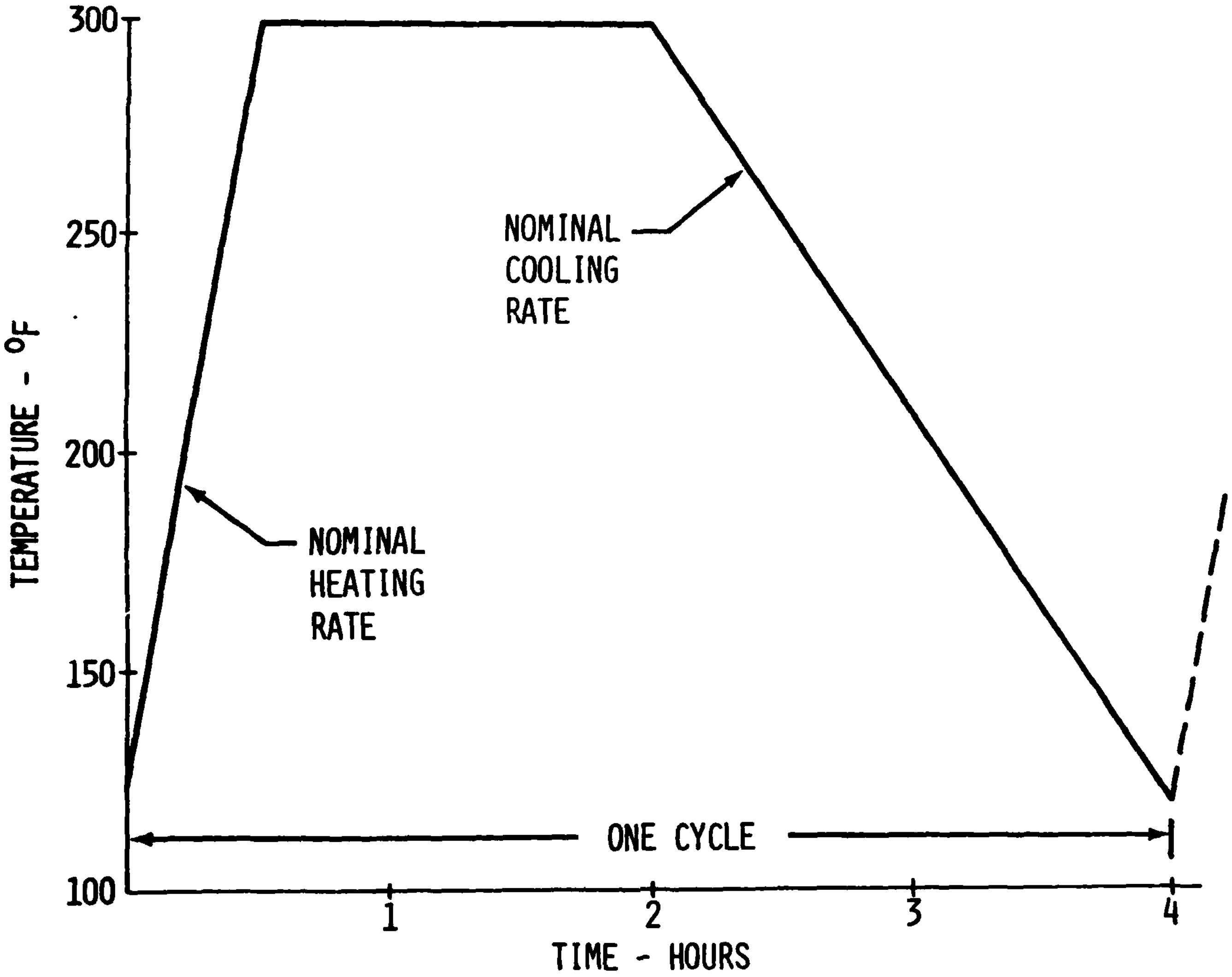

Figure 8. Valve stiction test temperature profile

TABLE 5 VALVE STICTION TEST RESULTS

Fluid	Max. Valve Slide Force	Fluid Acid No. Before/After	Viscosity, CS Before/After
E6.5	0.5 lb		5.2/6.0 (+15%)
AO-8	1.0 lb	.05/.05	7.3/70.0 (+859%)
MIL-H-5606	<0.1 lb	.02/.05	14.4/20.6 (+43%)
MIL-H-83282	<0.1 lb	.05/.01	15.7/15.8 (+0.6%)
MIL-H-27601	<0.1 lb	0.0/0.0	14.8/15.0 (+1.3%)
M2V	<0.1 lb	.22/.24	15.7/16.4 (+4.2%)

2.3.2.4 Fluid Stability and Metals Compatibility Tests

To determine the high-temperature stability of the candidate fluids and their relative compatibility with metals in a number of simulated system environments, a series of thermal, oxidative, and hydrolytic stability tests were run with the test fluid in contact with specimens of typical hydraulic component metals. These tests were performed by the Materials Laboratory with American Instrument Company (Aminco) rocking hydrogenation bombs. A stack of the following metal specimens was used in each test:

a. 52100 bearing steel ball
b. 4640 bronze disc
c. 3Al-2.5V titanium tube
d. 4340M steel disc
e. M-50 tool steel ball
f. 21Cr-6Ni-9Mn stainless steel tube
g. 440C stainless steel ball
h. 6061-T6 aluminum wafer
i. 15-5PH stainless steel disc
j. K6E cast iron ring
k. Nitralloy 135-M steel disc

In each of the stability tests, the bombs containing the test fluid and the metal specimens were heated to 325F and rocked about a pivot for 72 hours with the fluid sloshing over the metal specimens. For comparison, a thermal stability test was also run with MIL-H-83282 fluid at 450F (its rated maximum operating temperature); and, oxidative and hydrolytic stability tests were run with MIL-H-5606 fluid at 325F. In the thermal-stability-corrosion tests, an atmospheric blanket of 100-psig dry nitrogen was maintained in the bomb; and, in the oxidative-stability-corrosion tests, 100-psig air was maintained. In the hydrolytic-stability-corrosion tests, a 0.2 percent volume concentration of water was mixed with the test fluid and 100-psig dry nitrogen atmosphere was maintained.

Following those tests, fluid properties were measured to determine the degree of breakdown; and, the metal specimens were weighed to determine the degree of corrosive attack or oxidation. The results presented in Table 6 indicated that the AO-8 fluid suffered no significant breakdown. The weight change of all metal specimens in the AO-8 fluid was within the target limits of ± 0.20 mg/cm^2 except the 4640 bronze, which exceeded the limit in all three tests, and the 52100 bearing steel, the M-50 tool steel, and the 15-5PH

TABLE 6 FLUID STABILITY AND METALS COMPATIBILITY TEST DATA

PARAMETER	TARGET VALUE	THERMAL-STABILITY TEST DATA			OXIDATIVE-STABILITY TEST DATA			HYDROLYTIC-STABILITY TEST DATA		
TEST CONDITIONS TEMPERATURE		72-Hr With Dry Nitrogen 325F	325F	450F	72-Hr With Dry Air 325F	325F	325F	72-Hr, N_2 With 0.2% H_2O 325F	325F	325F
FLUID PROPERTY CHANGE		AO-8 CTFE	E6.5 Freon	MIL-H 83282	AO-8 CTFE	E6.5 Freon	MIL-H -5606	AO-8 CTFE	E6.5 Freon	MIL-H -5606
Neutralization No.	≤ 0.2	<0.1	<0.01	0.64	<0.1	<0.1	0.2	<0.1	<0.1	<0.1
Viscosity @ 100F	$\leq \pm 5\%$	-2.4	+0.6	-1.0	-1.9	+7.0	-1.0	-2.3	-.01	+1.1
METAL WT. CHANGE	$\leq \pm 0.20$ mg/cm^2									
52100 Steel Ball	"	0.00	+0.02	-0.34	0.00	+0.15	+0.13	-0.22	-0.06	-0.56
4640 Bronze Disc	"	-0.22	0.00	+0.60	+0.30	+0.28	-0.08	+0.54	+0.11	+0.05
3Al-2.5V Ti Tube	"	-0.01	0.00	-0.02	-0.03	+0.01	0.00	-0.03	-0.03	-0.01
4340M Steel Disc	"	+0.06	-0.01	-0.04	-0.01	+0.14	+0.07	-0.09	-0.02	-0.08
M-50 Steel Ball	"	+0.01	-0.05	-0.44	+0.01	+0.16	+0.13	-0.25	-0.02	-0.45
21-6-9 Steel Tube	"	+0.02	-0.01	0.00	+0.01	+0.01	+0.04	-0.05	0.00	-0.02
440C Steel Ball	"	+0.01	-0.04	0.00	-0.04	+0.01	+0.05	-0.04	+0.02	-0.01
6061-T6 Al Wafer	"	+0.01	-0.04	-0.01	-0.02	+0.04	-0.02	-0.01	+0.01	-0.04
15-5PH Steel Disc	"	+0.01	0	+0.02	+0.01	+0.03	+0.02	-0.28	-0.01	-0.01
K6E Cast Iron Ring	"	0.00	-0.04	-0.18	0	+0.03	-0.01	-0.07	+0.01	+0.03
Nitralloy Steel Disc	"	+0.02	-0.01	-0.02	0	+0.04	+0.05	-0.09	+0.03	-0.14

stainless steel which had weight losses slightly in excess of the limit in the hydrolytic stability test. The E6.5 fluid showed no indication of fluid breakdown other than a gain in viscosity which slightly exceeded the target limit in the oxidative stability test. No excessive material reactions were found except for the 4640 bronze which was slightly above the target weight gain limit in the oxidation stability test.

In the tests of the two reference fluids conducted to confirm the target values, it was found that, during the thermal stability test of the MIL-H-83282 fluid, its neutralization number exceeded the target limit. During that test, the weight change of the 4640 bronze and the 52100 and M-50 steels exceeded the target limit somewhat more than the comparable changes during the hydrolytic stability test of the AO-8 fluid. During the hydrolytic stability test of the MIL-H-5606 fluid, the weight loss of the 52100 and M-50 steels also exceeded the target limit by similar amounts. The data resulting from these tests are also shown in Table 6.

The Materials Laboratory also conducted the Federal Standard 791B Corrosiveness and Oxidation Stability Test per Method 5308 at 275F. The E6.5 and AO-8 fluids passed the MIL-H-5606 requirements. However, in the copper strip corrosion test, the AO-8 fluid failed to meet the maximum tarnish/corrosion requirements per MIL-H-5606. As with the Aminco bomb tests of the 4640 bronze, the copper strip tended to react with the AO-8 fluid indicating that there may be a problem with the use of copper bearing alloys.

2.3.2.5 Elastomer Compatibility Tests

In support of the efforts directed towards the development of a nonflammable hydraulic system, the Fluids, Lubricants & Elastomers Branch of the AFWAL Materials Laboratory initiated a program to develop elastomeric seals compatible with the main nonflammable fluid candidates - Halocarbon AO-8 and Freon E-6.5. Many off-the-shelf seals and compounds based on new experimental elastomers as well as most commercially available elastomers were screened in seeking seals that were both chemically and physically compatible with each of these candidate fluids. Details of this seal development effort including dynamic testing are covered in a separate report, Reference 4. In summary, the results of this rudimentary effort showed that seals based on phosphonitrilic fluoroelastomer (PNF) and ethylene propylene diene monomer (EPDM) rubber to be the most suitable for use with respectively the Halocarbon AO-8 and Freon E-6.5 candidate hydraulic fluids. Upon the recommendation of AFWAL's Materials Laboratory, seals based on an experimental Firestone PNF compound (Shore A Hardness 67) were used for assessing the dynamic performance capabilities of the prime candidate nonflammable fluid - Halocarbon AO-8.

2.3.3 Predicted Impact on Hydraulic System Components

Prior to the selection of one fluid for further evaluation, estimates of the comparative impact which the two candidate fluids would have on the performance of the following hydraulic components, and the design

4. T. L. Graham and W. E. Berner, Development of Seals for Nonflammable Hydraulic Fluids, AFML-TR-79-4143, Air force Materials Laboratory, January 1980.

changes which would be required to obtain performance equivalent to that obtained with mineral fluid per MIL-H-5606, were made:

a. Hydraulic pumps
b. Hydraulic seals
c. Electrical insulation
d. Flight control servoactuators
e. Hydraulic fluid reservoirs
f. Heat exchangers
g. Pressure and flow control valves
h. Hydraulic filters

Existing test data were reviewed, analytical studies were made, and component suppliers were consulted. The findings are summarized as follows.

2.3.3.1 Hydraulic Pumps

A query was sent to the leading hydraulic pump suppliers requesting their estimate of the design changes which would be required to utilize either of the two candidate nonflammable fluids. Three manufactures responded with a variety of comments which, in general, confirmed the anticipated design change areas. Aero Hydraulics indicated that density would have "(no) significant effect on pumps and motors," and Abex stated; "higher density of the proposed fluids does not require pump design changes." Vicker's, on the other hand, noted that "flow into and out of a hydraulic pump can be considered as turbulent (therefore) flow losses in a pump or motor will be of the same magnitude", and "density ..... will cause increased power loss and lower efficiency." They also indicated that "due to increased fluid mass, valve plate erosion problems may be evident (if a) higher inlet pressure is not provided." Boeing agreed with the Vickers' statements but did not believe they would be of sufficient magnitude to cause any significant design or weight penalties.

All three respondents were concerned about the high-temperature viscosity of the candidate fluids. Each felt that the low viscosity at maximum pump case temperatures may cause lubricity failures of bearings, piston shoes, etc. Based upon Shell Four-Ball testing and absolute viscosity values, minimal differences should be expected. However, as with any fluid, a pump must be to some extent tailored to that fluid.

Both Abex and Vickers indicated that lower pumping efficiencies would be attained with the lower bulk modulus values. The volumetric pumping losses due to fluid compression for the E6.5 fluid would be double the losses for the MIL-H-5606 fluid, but the increase in the physical size of a pump (to compensate for volumetric inefficiency) would be insignificant. Fluid compression losses at 3000 psi and 240F would be 2.25% for the AO-8 fluid, and 4.2% for the E6.5 fluid, compared to 2% for the MIL-H-5606 fluid.

The higher -65F viscosity of the candidate fluids relative to MIL-H-5606 was a concern of both Abex and Vickers. Abex stated that the "high viscosity at -65F may pose problems due to greatly increased viscous drag forces on the shoes and hold-down mechanism during the inlet stroke." However, Vickers commented that "fluid shear is calculated from the equation $T = V\mu/t$ where V is velocity, μ is absolute viscosity, and t is the thickness

of the oil film." Fluid shear is then an indication of the fluid drag force, which at -65F would be 150% greater for the A0-8 fluid, and 200% greater for the E6.5 fluid.

Vickers also stated that "neither ... fluid appear(s) to be suitable for operation at -65F." Boeing disagreed with this latter statement because of experience with cold system startup testing conducted during the SST program with the very viscous polyol ester hydraulic fluid. The test system consisted of a modeled SST hydraulic system with all pertainent physical features being accounted for. A summary of the results is presented in Table 7. Considering the much greater viscosities successfully pumped, the candidate fluids appear to be suitable for operation at -65F.

TABLE 7 PUMP COLD-START PERFORMANCE WITH A HIGHLY VISCOUS HYDRAULIC FLUID

Initial Sys. Temperature (Deg. F)	Fluid Viscosity (cs.)	Time to Sustained Pressure (min.)	Time to Full Flow (min.)
-34	4,900	0.1	2.5
-42	8,000	0.7	2.7
-44	9,200	0.7	4.1
-48	12,000	3.7	5.2
-52	16,500	0.7	4.8
-54	19,500	0.7	6.5
-57	25,000	0.5	6.0
-63	42,000	8.8	9.4

Fluid: Humble Oil ETO 5251, trimethylolpropane ester Type 2 jet engine oil

Pump: Abex AP10V 1.77-cipr variable-displacement inline piston pump

Suction Line: 40 ft, 1-1/4 line from 60 psig reservoir

Load: Relief valve set for 2500 psid

Pump Speed: Startup and run at 3540 rpm (scaled idle speed)

Each test run conducted for the SST program was continued until the system warmed up to 60F. A pump teardown inspection was conducted after each run, and after reassembly a running performance check was made. These inspections noted no change in case-drain flow and only slight cavitation erosion pitting visible in the barrel kidney port area.

As part of the candidate nonflammable fluid evaluation, the Materials Laboratory conducted a 100-hour pump loop test on each candidate fluid in the test system shown in Figure 9. A New York Air Brake Stratopower aircraft hydraulic pump was used to provide 1.5 to 5 gpm cyclic flow at 3000 psi. The pump inlet temperature was regulated to a constant 250F, with the typical pump discharge flow temperature running near 275F.

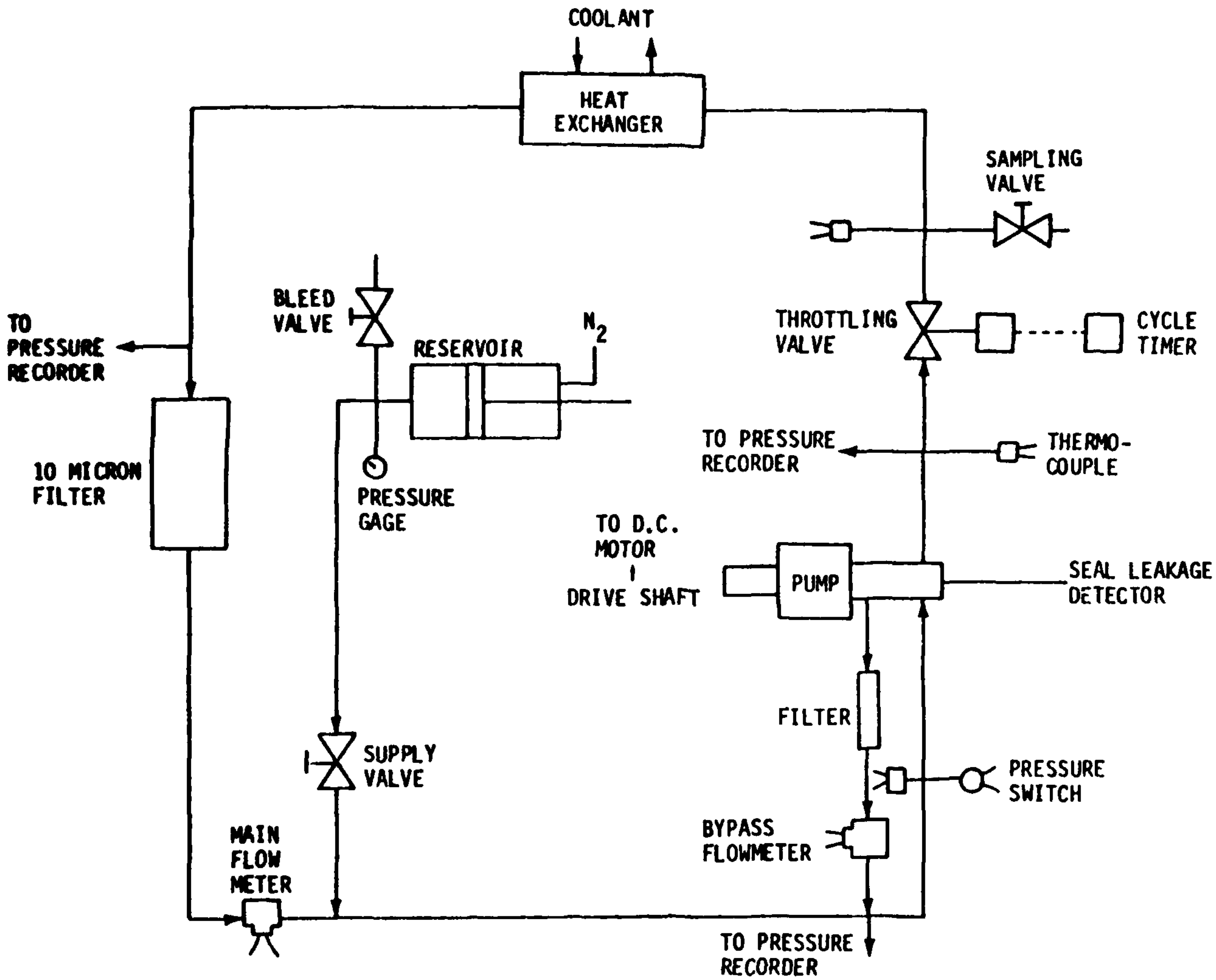

Figure 9. Pump loop test circuit

Both the AO-8 and E6.5 fluids showed no signs of degradation following the pump loop testing. The pump was disassembled for inspection following each test. The E6.5 pump contained some wear in the check valve stem guides but it was emphasized that these were rebuilt pumps and that the wear may have been there prior to testing. The high-load high-velocity parts such as the piston slippers, cam plate, and bearings had only normal wear. The AO-8 pump also had only normal wear patterns on the high-load high-velocity pump parts and somewhat less wear in the check valve stem guides. Again, this guide wear may have been in evidence at the start of testing.

Therefore, based upon the above data and information, it was concluded that either fluid could be pumped with conventional piston pumps, and that the effect on performance and any required design changes could be determined by testing a more modern pump design, at its rated conditions, during component compatibility testing later in the program.

2.3.3.2 Hydraulic Seals

One of the most important criteria in evaluating a hydraulic fluid is the ability to contain it in a system with inexpensive elastomeric seals, as opposed to the more expensive plastic and metallic seal designs. Contemporary hydraulic components use a large number of seals; and, the most cost-effective approach is to utilize elastomeric O-rings in the standard sizes for most seal applications.

As noted in Section 2.3.2.5, the seal tests conducted by the Materials Laboratory indicated that two materials, a phosphonitrilic fluorelastomer (PNF) and an ethylene propylene diene monomer (EPDM) rubber were the most suitable materials for use with respectively the Halocarbon AO-8 and Dupont Freon E6.5 fluids.

2.3.3.3 Electrical Insulation

Each candidate fluid's compatibility with electrical system insulation materials was also studied. DuPont provided information indicating that several commonly used insulation materials are compatible. They also presented compatibility data for similar "Freon" E homologs showing no deterioration of many insulation materials at high temperature for a reasonable length of time. No compatibility information for AO-8 fluid was made available other than Halocarbon's statement that "silicone rubbers are not compatible" and that other commonly used elctrical insulation materials ... should be compatible."

Due to the inertness (non-solubility) of both candidate fluids with many plastics and elastomers, it was concluded by Boeing and the Materials Laboratory that several standard electrical insulations and potting compounds capable of 300F environments are available for either candidate.

2.3.3.4 Flight Control Servoactuators

A determination of the effect of each candidate fluid upon the performance of a typical flight control servoactuator was made by reactivating the B-52G/H elevator actuation system analog computer simulation at the Boeing

Wichita Plant. Density and bulk modulus values for the candidate fluids were substituted for the MIL-H-5606 fluid values, and the actuator response characteristics computed.

Fluid density affects the valve flow and pressure drop per the relationship:

$$\frac{Q^2}{\Delta P} = \frac{KA^2}{\rho} \qquad (1)$$

where:

Q = valve flow rate
ΔP = valve pressure drop
K = a constant
ρ = fluid density
A = valve metering slot area

Therefore, if valve gains (e.g. no-load flow rates) equivalent to that obtained with MIL-H-5606 fluid are desired with the higher density fluids, the valve slot areas may be increased for density variation through the formula:

$$\frac{A_2}{A_1} = \left(\frac{\rho_2}{\rho_1}\right)^{.5} \qquad (2)$$

For the candidate fluids, with densities of approximately 1.83, compared with 0.84 for MIL-H-5606 fluid, the B-52 elevator valve slot widths could be increased from .062 to .092 inches to obtain equivalent gain.

The fluid modulus has a primary affect upon the servoactuator's response to an oscillation at the valve input and actuator output. A servoactuator specification will generally require an exacting match between the command and output for oscillatory inputs up to those rates required for aircraft stability and/or maneuvers, and will require that the output:command-input ratio be greatly reduced when approaching the aerodynamic flutter and the structure/actuator loop spring natural frequencies. Reduced fluid bulk modulus, as is the case with both candidate fluids, has the effects of slightly reducing the higher frequency output:command-input ratio and reducing the structure/actuator loop spring natural frequency. The former effect is a slight attribute but the reduced natural frequency effect decreases the margin between an acceptable actuator amplitude-ratio/frequency and the structure/actuator spring loop natural frequency.

Since the worst case analysis is the usual method for determining the dynamic condition acceptability of an actuator installaation, the maximum fluid temperature is used. Bulk modulus is lowest at maximum temperature producing the softest fluid spring and the lowest surface induced natural frequency. All lower fluid temperature have greater bulk moduli, therefore

responding better at higher frequencies. The bulk moduli of the candidate fluids never exceeds that of MIL-H-5606 at corresponding temperatures thus their amplitude-ratio values will not exceed those of MIL-H-5606.

Amplitude ratio versus frequency curves for the candidate fluids and for MIL-H-5606 fluid are shown in Figure 10. Because the candidate fluids have a higher density, the existing control valve will allow a no-load actuation rate of 54 degrees per second for the candidate fluids (subscript 2) as compared to 80 degrees per second for MIL-H-5606 fluid (subscript 1). This is calculated from the equation:

$$\omega_2 = \omega_1 \left(\frac{\rho_1}{\rho_2}\right)^{.5} \quad \text{...} \quad (3)$$

as derived from: $$Q = KC\ lw \left(\frac{\Delta P}{\rho}\right)^{.5} \quad \text{...............................} \quad (4)$$

where $Q = AV$ and $V = r\omega$

To compare the performance of the candidate fluids with MIL-H-5606 fluid in the same installation:

$r_1 = r_2$, $A_1 = A_2$, $K_1 = K_2$, $C_1 = C_2$, $l_1 = l_2$, $w_1 = w_2$ and $\Delta P_1 = \Delta P_2$

$$\frac{Q_2}{Q_1} = \frac{A_2\ r_2\ \omega_2}{A_1\ r_1\ \omega_1} = \frac{K_2\ C_2\ l_2\ w_2}{K_1\ C_1\ l_1\ w_1}\left(\frac{\Delta P_2 \cdot \rho_1}{\Delta P_1 \quad \rho_2}\right)^{.5} \quad \text{....................} \quad (5)$$

$$\text{and} \quad \frac{\omega_2}{\omega_1} = \left(\frac{\rho_1}{\rho_2}\right)^{.5} \quad \text{...} \quad (6)$$

where:

A	=	actuator piston area
C	=	valve discharge coeffiecient
K	=	constant
l	=	valve slot length in stroke direction
ΔP	=	valve pressure drop
Q	=	valve flow rate
r	=	surface/actuator moment arm
V	=	actuator piston velocity
w	=	valve slot width
ρ	=	fluid density
ω	=	surface angular velocity

If it is desired that the no-load actuation rate remain at 80 degrees per second for the candidate fluids as it was with MIL-H-5606 fluid, and the major physical design features of the servo actuator remain unchanged, then $A_1 = A_2$, $r_1 = r_2$, $K_1 = K_2$, $C_1 = C_2$, $l_1 = l_2$, $\Delta P_1 = \Delta P_2$ and $\omega_2 = \omega_1$. Equations (5) and (6) hold, and the valve slot width should be increased as follows:

$$w_2 = .062 \left(\frac{1.83}{.84}\right)^{.5} = .092$$

As shown in Figure 10, the response curves for the candidate fluids with the existing valve slot width (.062) vary significantly from the MIL-H-5606 curve. However, with the slot width increased (to .092) to provide an equivalent flow gain, the curves nearly coincide. The specification limits are also included in Figure 10.

Figure 11 presents the phase lag between a sinusoidal input command and the actuator output for the candidate fluids and for MIL-H-5606 fluid. The phase lag indicates the sinusoidal angle or period of time that the output follows the input command. No delay is desirable to attain good rate response. Again the candidate fluid curves deviate substantially from the MIL-H-5606 fluid curve because of the density, but nearly coincide when the valve slot width is opened up to give the same no-load rate.

The dual-tandem servoactuator is required to remain stable with only one piston active, thus the frequency response performance for that condition was also examined. As shown in Figures 12 and 13 respectively, the amplitude ratio and phase lag curves for the candidate fluids (with the .092 valve slot widths) compare very closely with the curves for MIL-H-5606 fluid (with a .062 slot width) both for dual and single cylinder operation.

It was concluded that, the B-52 elevator servoactuator's response would be acceptable with either candidate fluid when modified with larger valve slot widths. However, it should not be concluded that all aircraft servoactuators can be modified as easily to obtain satisfactory performance with the candidate fluids.

2.3.3.5 Hydraulic Fluid Reservoirs

Reservoirs are often the largest and heaviest single components in a hydraulic system; and, the system designers may experience considerable difficulty in finding a satisfactory location for their installation. The following fluid properties should be considered by the reservoir designer.

a. Thermal coefficient of expansion

Sufficient reservoir volume must be provided to accommodate the expansion of the total system fluid volume from the minimum

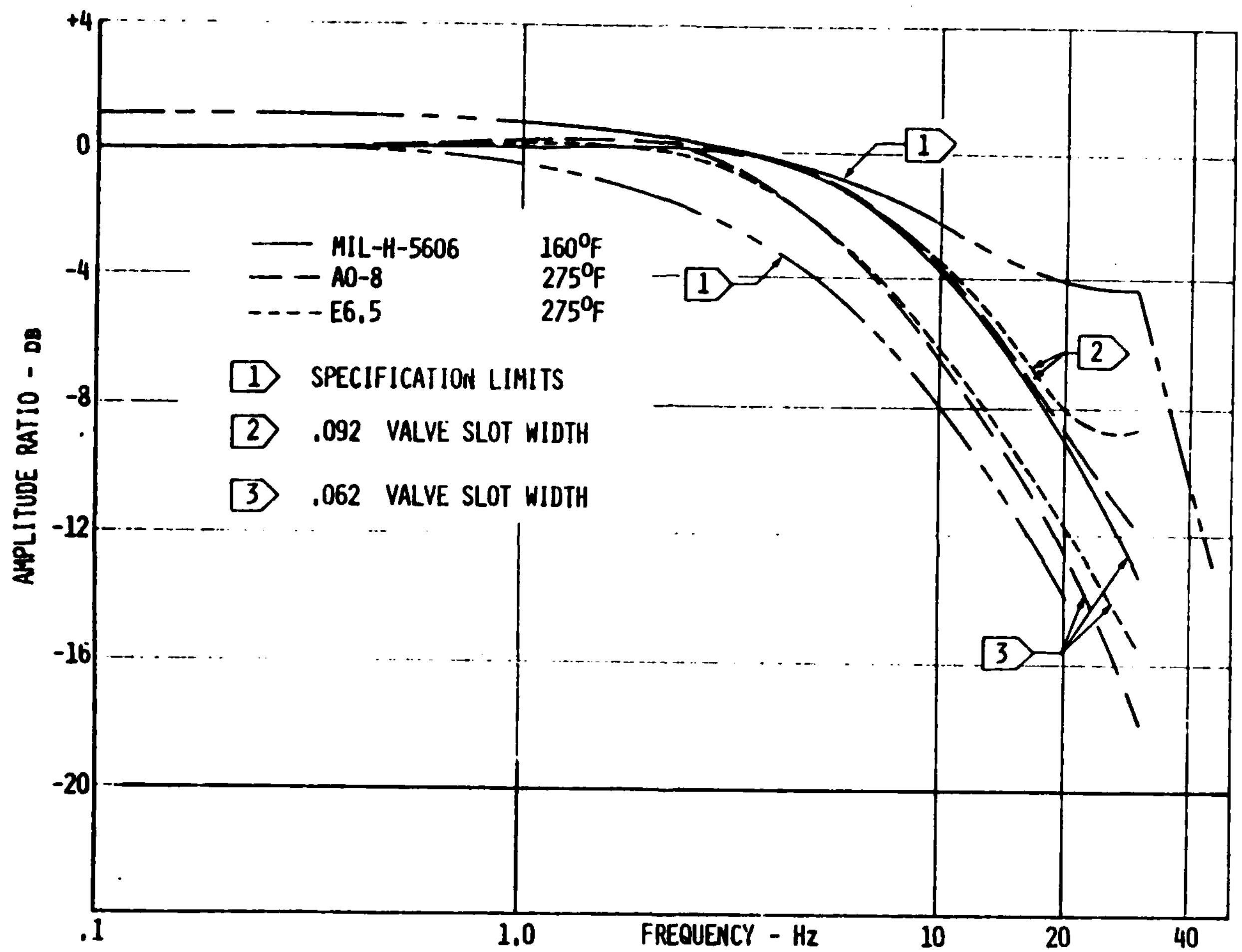

Figure 10. B-52 elevator system frequency response (amplitude ratio) with the two candidate fluids and with MIL-H-5606 fluid

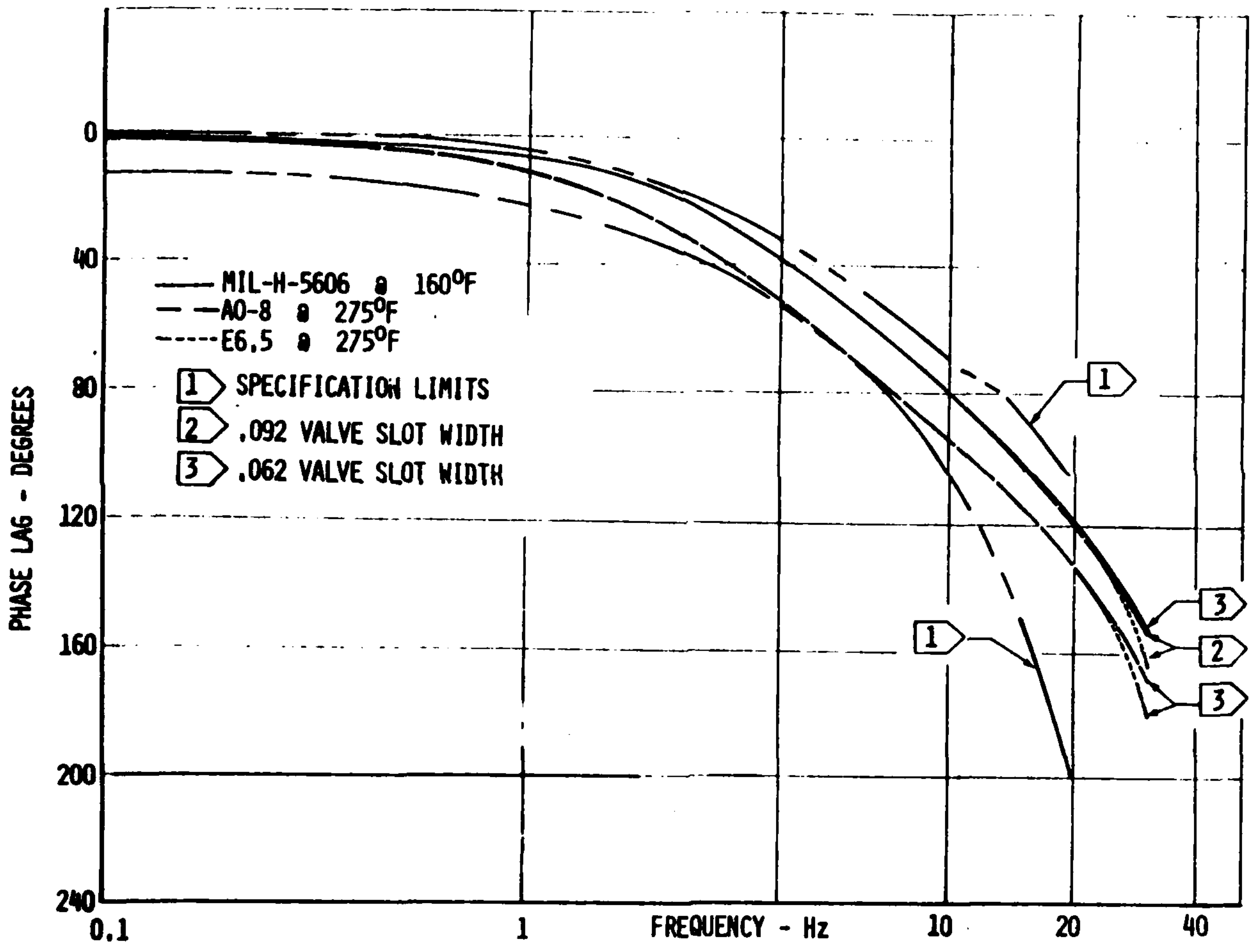

Figure 11. B-52 elevator system frequency response (phase lag) with the two candidate fluids and with MIL-H-5606 fluid

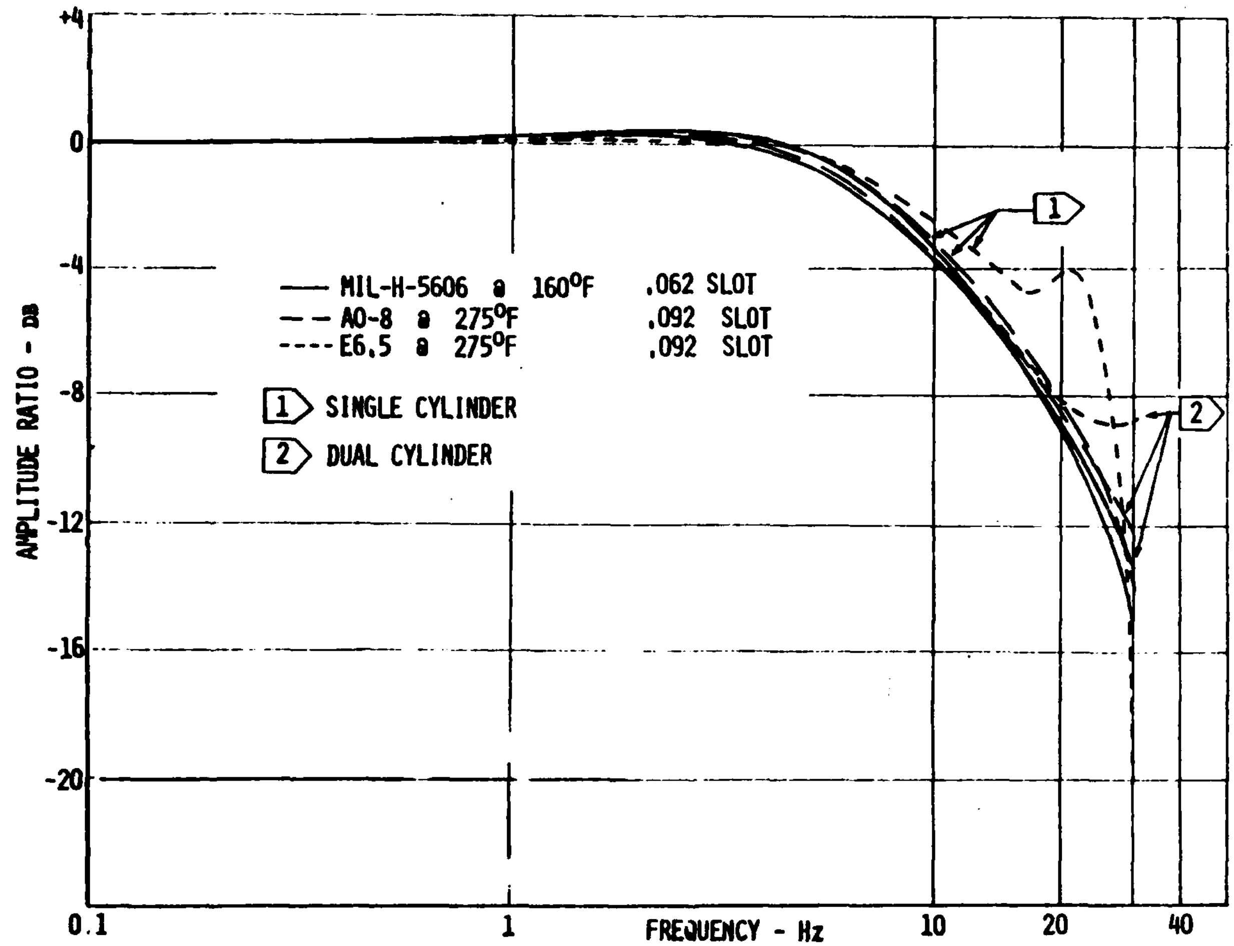

Figure 12. Comparison of B-52 elevator system frequency response (amplitude ratio) with dual-system and single-system operation

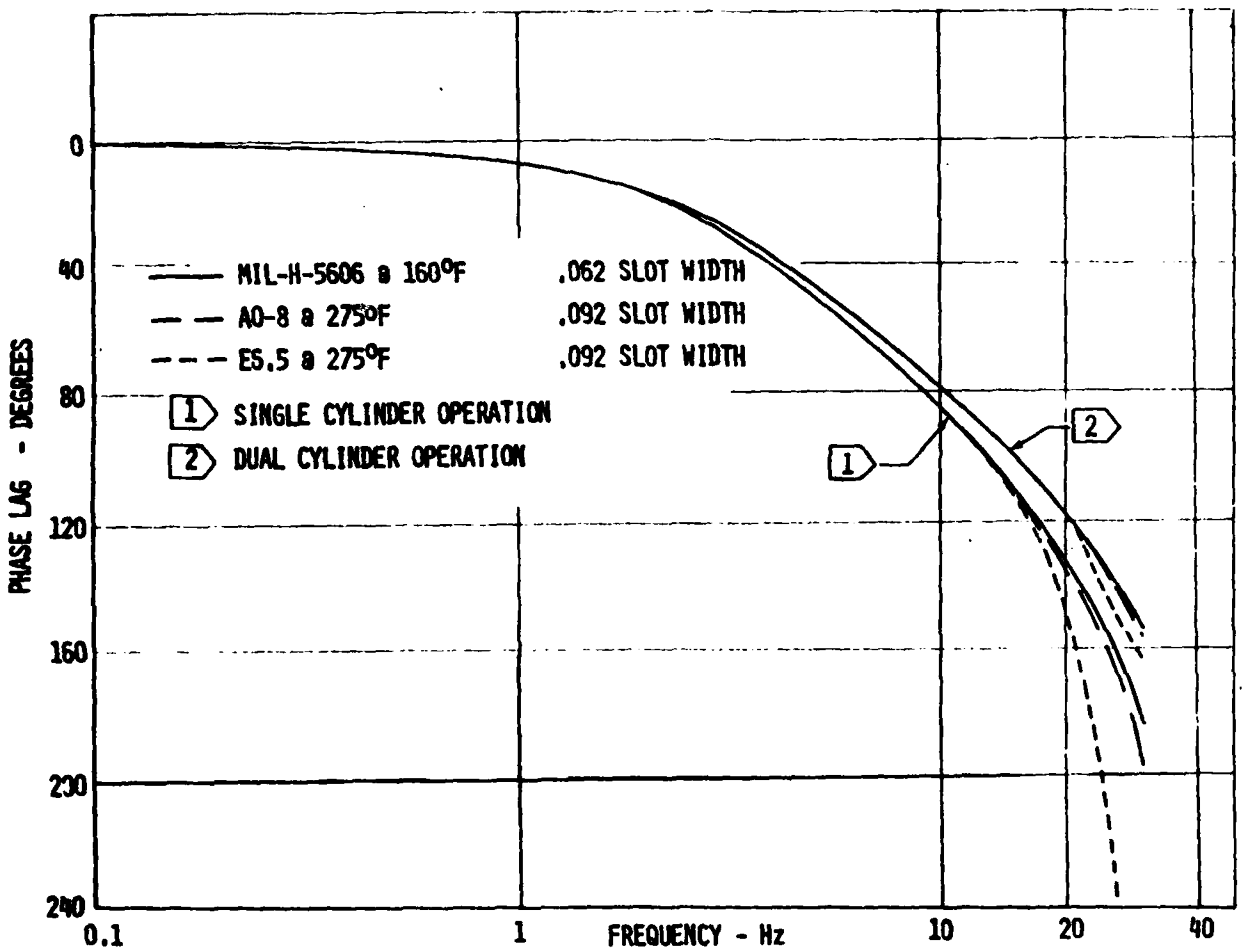

Figure 13. Comparison of B-52 elevator system frequency response (phase lag) with dual-system and single-system operation

design temperature to the maximum design temperature, e.g. from -65F to 275F for a Type II system.

b. Bulk modulus

Sufficient fluid volume must be provided in the reservoir to accommodate the fluid compressed in the system pressure lines and components.

c. Vapor pressure

The reservoir pressurization pressure must be high enough to prevent fluid vaporization.

d. Bubble collapse rate

A low collapse rate could indicate a foaming tendancy which could dictate the use of a separated type reservoir design.

Other fluid properties that indirectly affect the reservoir size are viscosity and density as they affect system volume through line sizing.

In Table 8, the total system fluid volumes and the total reservoir fluid thermal expansion volumes required for MIL-H-5606 fluid and the two candidate fluids are shown for three aircraft types. The thermal expansion volumes shown would indicate that reservoirs for the candidate fluids should be nearly 20 to 25% larger than for MIL-H-5606 fluid. However, since the total reservoir capacity is generally two to three times that of the expansion volume, the actual increase required would be closer to ten percent.

The fluid volume required in the reservoir to make up volume compressed in the pressure manifold (pressure lines, etc.) when the system is pressurized to rated pressure is a function of the fluid's bulk modulus. Since the bulk modulus is the inverse of the fluid compressibility, the differential volume per unit volume is expressed by the equation:

$$V = \frac{P}{\beta} \quad \text{..} \quad (7)$$

For a 3000 psi system at room temperature with MIL-H-5606 fluid:

$$\Delta V = \frac{3000}{273{,}300} = .010977 \text{ or } 1.1\%$$

and with AO-8 fluid:

$$\Delta V = \frac{3000}{240{,}700} = .012464 \text{ or } 1.2\%$$

TABLE 8 RESERVOIR THERMAL EXPANSION VOLUME

AIRCRAFT	TOTAL FLUID VOLUME - gal.	TOTAL THERMAL EXPANSION VOLUME - gal.		
		MIL-H-5606	AO-8	E6.5
C-14A	130	11.7	14.6	14.0
F-15	23	3.1	3.9	3.7
F-16	12	1.6	2.0	2.0
COEF. OF THERMAL EXPANSION - $in^3/in^3/°F$		.00040	.00050	.00048

TABLE 9 RESERVOIR FLUID COMPRESSIBILITY VOLUME

AIRCRAFT	TOTAL FLUID VOL - gal.	PRES. MANIFOLD VOL - gal.	COMPRESSION VOLUME - gal. MIL-H-5606	AO-8	E6.5
C-14A	130	39	0.4	0.5	0.7
F-15	23	6.9	0.08	0.08	0.13
F-16	12	3.6	0.04	0.04	0.07
BULK MODULUS ADIABATIC TANGENT (psi) at 77F			273,300	240,700	157,300

and with E6.5 fluid:

$$\Delta V = \frac{3000}{157,300} = .019072 \text{ or } 1.9\%$$

As can be noted from Table 9, the effect of fluid compressibility (bulk modulus) upon reservoir size is minimal.

A fluid's vapor pressure is directly related to the critical inlet pressure of a hydraulic pump. A pump's critical inlet pressure is the minimum inlet pressure below which cavitation commences. Thus a fluid with a high vapor pressure will require a higher reservoir pressure than a low vapor pressure fluid. Although other temperature conditions may also require examination depending upon the suction line, pump, and flight temperature profile parameters, the maximum temperature condition should always be evaluated. The maximum pump inlet temperature for a Type II system fluid temperature of 275F would be approximately 225F.

As shown in Table 10, these vapor pressures indicate very small differences in comparison to the significantly greater reservoir pressure required to ensure flow through the pump suction line and adequate pressure at the pump inlet port (usually 30 to 50 psi). Therefore, no weight or design effects would be anticipated.

TABLE 10 FLUID VAPOR PRESSURE @ 225F

Fluid	Vapor Pressure
MIL-H-5606	13.5 mm of Hg = .26 psi
AO-8 CTFE	10.0 mm of Hg = .193 psi
"Freon" E6.5	4.2 mm of Hg = .08 psi

Reservoirs designed for either of the candidate fluids would be affected primarily by the thermal coefficient of expansion resulting in a larger capacity and higher weight.

A detriment to many systems in the past is a fluid's tendancy to form foam when entrained and dissolved air form free air bubbles. If large volumes of foam develop in the distribution system, the fluid volume displaced must be accommodated in the reservoir or dumped overboard. Foam entering a hydraulic pump inlet line can have disasterous effects upon lubrication and output flow. The design of the fluid inlets and outlets of a reservoir (especially a non-separated type) can have a great deal to do with foam generation. Separated reserviors don't entirely eliminate the foaming problem, however, as air may enter the system through system seals and failed component changes.

Each of the candidate fluids were tested for foaming tendency by the

Materials Laboratory; and, since they both passed the MIL-H-5606 requirements, this concern was minimized.

2.3.3.6 Heat Exchangers

Nearly all modern aircraft hydraulic systems require a heat exchanger to stabilize the maximum fluid temperature. Excessively high temperatures can rapidly deteriorate the hydraulic fluid as well as the elastomeric seals. The fluid properties most concerned with heat exchanger sizing (heat transfer area) are specific heat (heat capacity) and thermal conductivity.

A cursory view of the specific heat data showed that the values for the candidates are approximately half that of MIL-H-5606. The specific heat data supplied by the Materials Laboratory as well as comparative data for MIL-H-5606 is presented in Figure 4. Reviewing the equation for system heat capacity, it was found that with the doubling of density, the product results in nearly equal hydraulic system heat capacity.

$$Q = C\ m\ \Delta T \quad \text{.......} \quad (8)$$

where:

Q = quantity of heat, BTU
C = coefficient of specific heat or heat capacity, BTU/lb/^oF
m = mass, lb_m
ΔT = temperature differential, degrees Fahrenheit

Thus, if it is assumed that the comparison fluid's maximum system temperatures are equal, then

$$\frac{Q_2}{Q_1} = \frac{C_2}{C_1}\ \frac{m_2}{m_1} \quad \text{.......} \quad (9)$$

then, if Q and m are per unit volume, $\rho = \frac{m}{v}$ and $Q' = \frac{Q}{v}$ and

it follows that

$$\frac{Q_2'}{Q_1'} = \frac{C_2}{C_1}\ \frac{\rho_2}{\rho_1} \quad \text{.......} \quad (10)$$

At maximum system bulk fluid temperature of 275F, the ratio of heat capacities for the candidate fluid systems compared to a MIL-H-5606 system would be as follows:

for an AO-8 fluid system: $\frac{Q_2'}{Q_1'} = \frac{.255}{.55}\ \frac{1.68}{0.77} = 1.012$

for an E6.5 fluid system: $\frac{Q_2'}{Q_1'} = \frac{.296}{.55} \quad \frac{1.595}{0.77} = 1.115$

The other primary fluid property in heat exchanger design is the thermal conductivity. The thermal conductivity of the candidate fluids and MIL-H-5606 fluids are shown in Figure 5. Heat exchanger sizing generally follows the thermal conductivity coefficient when comparing fluids if their heat capacities per unit volume and fluid flow rates are similar. This may be shown from the equation:

$$\frac{\Delta Q}{t} = KA\Delta T \qquad (11)$$

where:

Q = quantity of heat, BTU

Δt = time period, hr.

k = thermal conductivity coefficient, Btu/hr/ft^2/^{o}F/ft

A = heat exchanger plate area, ft^2

ΔT = temperature differential, degrees Fahrenheit

Therefore, when comparing fluids, the time element, Δt, remains the same, as does the temperature, ΔT, since it is desired that the maximum system temperatures remain equal. Then,

$$\frac{Q_2}{Q_1} = \frac{k_2}{k_1} \frac{A_2}{A_1} \quad \text{or} \quad \frac{A_2}{A_1} = \frac{Q_2}{Q_1} \frac{k_1}{k_2} \qquad (12)$$

Since the heat capacity of the fluids doesn't vary appreciably (as shown previously) and the system heat generation is not expected to be significantly different, then,

$$\frac{A_2}{A_1} = \frac{k_1}{k_2} \qquad (13)$$

Since the thermal conductivities of the candidate fluids are approximately one-half that of MIL-H-5606 fluid, the size of a heat exchanger in a system utilizing either candidate fluid must be approximately twice the size required for a MIL-H-5606 fluid system.

2.3.3.7 Flow and Pressure Control Valves

For other hydraulic system components such as flow and pressure control valves, fuses, etc., the candidate fluid's density and erosion

tendancy are the properties primarily affecting the design. These components contain orifices as their critical design feature; and, orifice size is principally a function of the design flow rate, allowable pressure drop, and the fluid's density. The previous servoactuator valve-slot-width discussion details those changes required when designing for the candidate fluids relative to MIL-H-5606 fluid.

In valves subject to leakage from high to low pressure, some commonly used fluids (phosphate esters) have the potential to cause metal erosion of the metering lands or sealing edges. This erosion phenomenon in turn increases the leakage at such valves and the total system quiescent flow. If allowed to continue, the system response becomes sluggish and potentially dangerous from lack of controllability.

Boeing discovered that the valve erosion experienced with phophate ester fluids was due to an electrochemical corrosion mechanism. It was also determined that the electrical conductivities of those fluids causing the valve erosion damage laid in a particular range of values. Figure 6 shows the erosion-prevalent band and the relationship of the candidate fluids and other common fluids to it. In tests at Boeing on this program, it was found that both candidate fluids were sufficiently removed from the erosion-prevalent band to indicate that no electrochemical valve erosion potential exists. Further, Du Pont data for E6.5 fluid indicated an extreme remoteness from the erosion band.

2.3.3.8 Hydraulic Filters

Hydraulic system filters used to protect components from particulate contamination are generally of the pleated element type using a treated paper, woven wire, or combination thereof as the filtering medium. For a determination of the required medium's (element) surface area, the following Darcy equation for laminar flow through a multiplicity of straight, constant-diameter capillary passages should be used.

$$Q = \frac{K N \pi d^2 \Delta P}{A \mu T} \qquad (14)$$

where:

K = permeability constant
Q = hydraulic fluid flow
N = number of flow passages
d = capillary diameter
T = length of capillary
μ = absolute viscosity
ΔP = differential pressure
A = filter media area

For an equivalent system using a candidate fluid as compared to MIL-H-5606 fluid, K, Q, ΔP, T and d would remain equal and Nd^2 would be proportional to the element area. Then,

$$\frac{A_2}{A_1} = \frac{\mu_2}{\mu_1} \qquad (15)$$

where: A = effective filter element area

Thus the candidate fluids filters would be somewhat larger that the MIL-H-5606 fluid filters in an equivalent system and the actual increase would depend upon the design temperature.

It is difficult to assess the impact on filter life, but several fluid properties can have an effect upon the amount of contaminant generated by the fluid or system. The fluid generated particulate matter is the result of fluid decomposing thermally, oxidatively, catalytically, chemically and/or by mechanical shear. In all of the fluid degradation aspects, the candidate fluids excell as they are basically very stable, inert materials. The E6.5 fluid has an advantage in that regard since it has no additives in its formulation.

System generated contamination is the result of normal system fluid replenishment, component mechanic l wear, seal wear, and component replacement. Of these, only fluid replenishment, component mechanical wear, and seal wear are related to the fluid properties. Fluid replenishment contamination is controlled by the quality control required by the fluid specification, the filling techniques and equipment, and the filtration (if any) in the filling circuit. Most of the component generated mechanical wear particles originate within the hydraulic pumps. Seal and anti-extrusion device (backups, cap rings, etc.) wear, results in elastomeric and plastic pieces nibbled or torn away from the seal or anti-extrusion device.

No appreciable difference in contamination generation between the candidate fluids could be determined from the testing completed to date. However, it was expected that the E6.5 fluid would have greater endurance life (no fluid breakdown) thus a lower level of contamination generation. Since both candidates are better than MIL-H-5606 fluid in this respect, no larger filter would be required. It is even possible that for equivalent element life, the element area could be reduced or a lower absolute rating (finer filtration) could be used.

2.3.4 Impact on System Tube Sizing

The overall impact of the candidate fluids upon a complete aircraft hydraulic system may be described in terms of the predicted changes in operating performance, increases in weight and cost, and potential changes in reliability, maintainability, and safety. However, it is assumed that, in adopting a new fluid, no major reductions in performance could be tolerated and that each system must be designed to respond to the various demands placed on it and provide flow at rates necessary to meet the operational requirements irrespective of the type of fluid used. Therefore, assuming that system tubing runs would be designed to provide the specified performance, and that they would be sized as necessary to avoid performance reductions, a study was made to determine the comparative impact of each of the two candidate fluids on tubing sizes necessary to obtain system performance equivalent to that obtained with MIL-H-5606 fluid.

Hydraulic fluid transmission lines fall into three general classificaitons: pressure lines, return lines, and pump suction lines. Due to the higher density of the candidate fluids, and their higher low-temperature absolute viscosity, tube sizes somewhat larger than those used for MIL-H-5606 fluid systems would be required. However, if lower-viscosity formulations of the candidate fuids proved to be acceptable, smaller tube sizes could be used.

2.3.4.1 System Pressure and Return Lines

The choice of tubing sizes for pressure and return lines is usually based on trade studies which balance energy loss against cost and weight. Large diameter tubing will conduct the fluid with lower pressure loss than smaller sizes, but will cost and weigh more.

Basically, tubing must be large enough so that, at all design conditions, pressure losses will not prevent all actuators from meeting their load and rate requirements. Secondly, the sizes must be large enough to prevent harmful dynamic pressures due to high fluid velocities. MIL-H-5440 specifies that "peak pressure resulting from any phase of the system operation shall not exceed 135 percent of the main system, subsystem, or return system pressure when measured with electronic equipment, or equivalent." The following discussions of those factors include the relevant equations for quantitative calculations.

2.3.4.1.1 Pressure-Loss Analysis

The typical hydraulic transmission system operating pressure cycle is illustrated graphically in Figure 14 which depicts the following four main phases of the cycle:

(1) The rise in pressure across the system pump.
(2) The pressure loss in the system pressure lines.
(3) The differential pressure available for actuating loads.
(4) The pressure loss in system return lines.

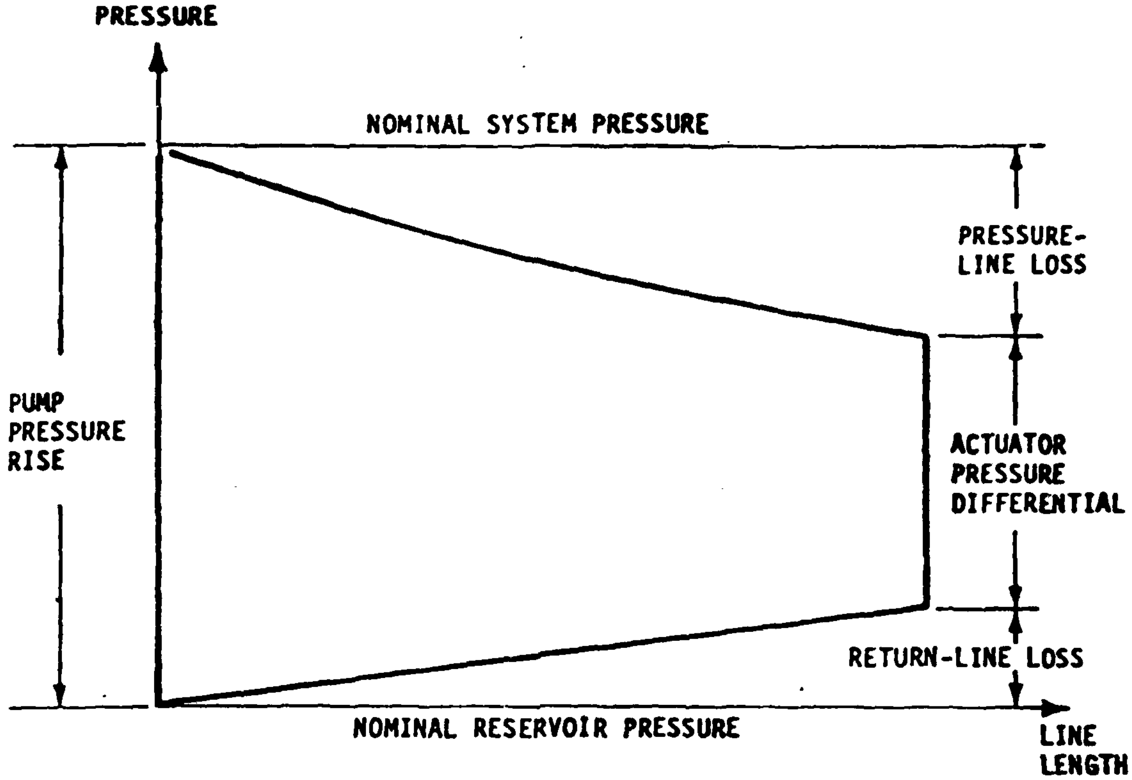

Figure 14 Typical hydraulic system pressure cycle

When the actuator loads, rates, and aircraft operating temperature requirements have been defined, the required hydraulic fluid flow rates and allowable pressure losses at various fluid operating temperatures can be established. From those requirements, and the necessary tube lengths for installation in the air vehicle, the required tube sizes can be calculated. This is generally done utilizing equations derived from the Darcy-Weisbach formula for lost head in round pipes, namely:

$$h_f = f \frac{L}{D} \frac{V^2}{2g} \quad \text{.......... (16)}$$

which was presented first in a somewhat more general form by Antoine Chezy in 1775. That formula has been reduced to the following form in order to utilize the generally used engineering units noted below:

$$P = 0.0135 \frac{fLsQ^2}{D^5} \quad \text{.......... (17)}$$

where:

- P = Pressure loss, pounds per square inch (psi)
- f = Friction factor, dimensionless
- L = Length of tube, feet (ft)
- Q = Fluid flow rate, US gallons per minute (gpm)
- D = Tube inside diameter, inches (in)
- s = Fluid specific gravity, dimensionless, or fluid density, grams per cubic centimeter (g/cm^3)

The friction factor (f) varies as a function of Reynolds number as shown in Figure 15 which can be found in several textbooks and other reference documents including the SAE Aerospace Recommended Practice ARP 24B (Reference 5). As can be seen in Figure 15, the relationship between friction factor and Reynolds number follows the following equations for laminar flow and turbulent flow respectively:

For laminar flow: $$f = \frac{64}{N_R} \quad \ldots\ldots\ldots\ldots \quad (18)$$

For turbulent flow: $$f = \frac{0.316}{N_R^{0.25}} \quad \ldots\ldots\ldots\ldots \quad (19)$$

As shown in ARP 24B, Reynolds number (N_R) is also a dimensionless factor which can be expressed in either of the following formulae:

$$N_R = \frac{\rho VD}{\mu} \quad \ldots\ldots\ldots \quad (20) \quad \text{or} \quad N_R = \frac{VD}{\nu} \quad \ldots\ldots\ldots \quad (21)$$

Substituting rate of fluid flow (Q) for fluid velocity (V), and specific gravity (s) for fluid mass density (ρ), results in the following forms:

5. SAE ARP 24B, Aerospace Recommended Practice, Determination of Hydraulic Pressure Drop, Society of Automotive Engineers, Inc., Warrendale, PA, 1-31-68.

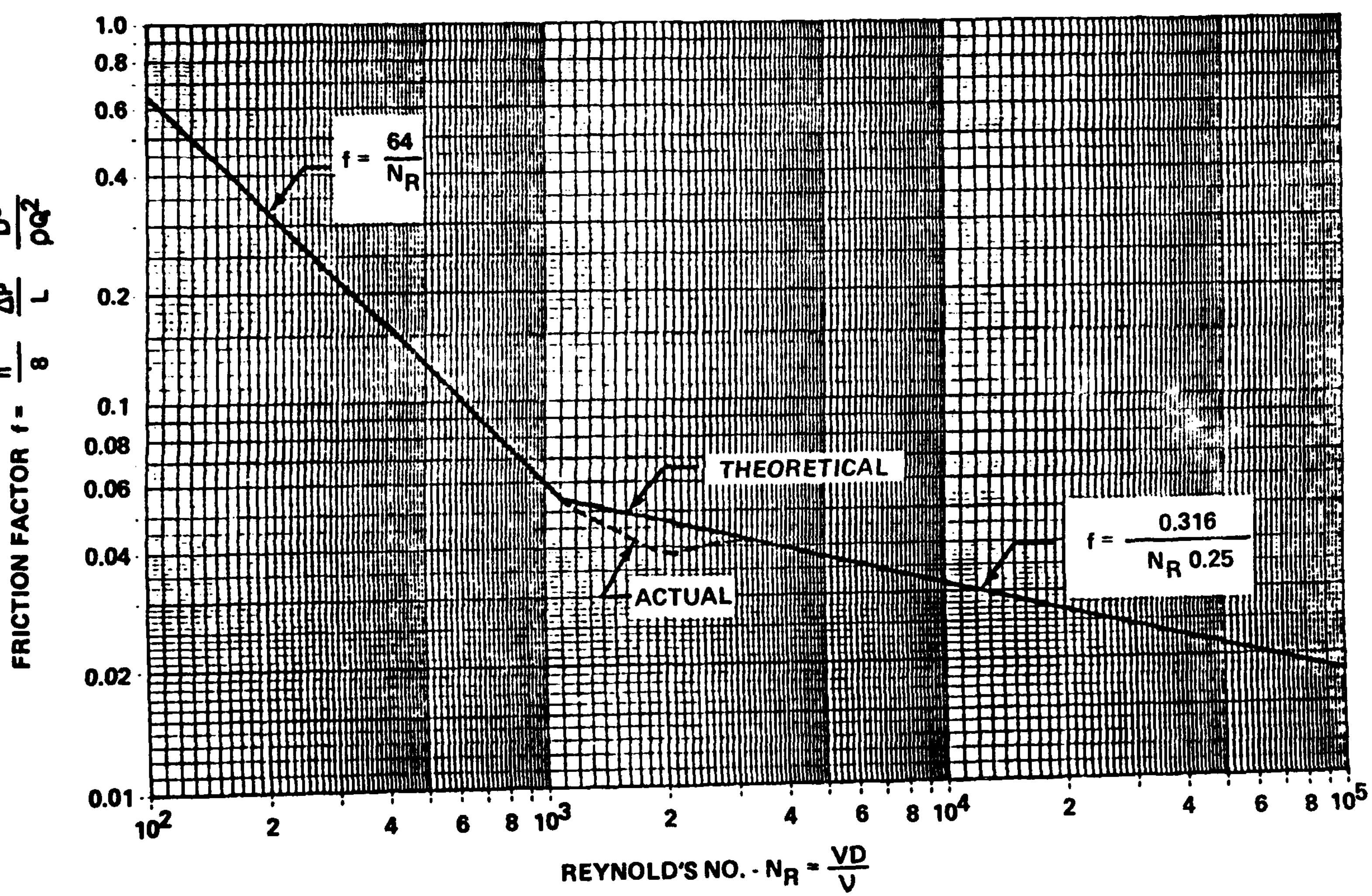

Figure 15. Friction factor f as a function of Reynolds number N_R for smooth tubes

$$N_R = 3160 \frac{sQ}{\mu D} \text{ .... (22)} \quad \text{or} \quad N_R = 3160 \frac{Q}{\nu D} \text{ .... (23)}$$

where:

- s = Fluid specific gravity, dimensionless, or fluid density, grams per cubic centimeter (g/cm^3)
- Q = Fluid flow rate, US gallons per minute (gpm)
- D = Tube inside diameter, inches (in)
- μ = Fluid absolute viscosity, centipoises (cp)
- ν = Fluid kinematic viscosity, centistokes (cs)

In general, for straight tubing, laminar flow is predominant for values of N_R below about 1,400, and becomes fully turbulent above about 3,600 (see Figure 15). When the flow is disturbed by the presence of bends and fittings, a turbulent condition is found to prevail down to N_R of 1,000 or less.

It is generally recognized that, under normal operating temperatures, high flow demands in aircraft hydraulic systems generally result in turbulent flow. However, tubing sizes are almost always determined by the requirement to keep pressure losses within established allowable limits under low-temperature conditions before the fluid has warmed up to its normal operating temperature. Under such conditions, flow is almost always laminar.

The friction factors for laminar flow and turbulent flow can be calculated from the following equations which were derived by substituting Eq. (22) into Eq. (18) and Eq. (19) respectively:

$$\text{For laminar flow:} \quad f = 0.02025 \frac{\mu D}{sQ} \text{ .... (24)}$$

$$\text{For turbulent flow:} \quad f = 0.042 \left(\frac{\mu D}{sQ}\right)^{0.25} \text{ .... (25)}$$

The laminar-flow and turbulent-flow pressure losses in tubing runs can be calculated from the following equations which were derived by substituting the foregoing friction factors into Eq (17).

$$\text{For laminar flow:} \quad \Delta P = 0.000{,}273 \frac{\mu Q L}{D^4} \text{ .... (26)}$$

$$\text{For turbulent flow:} \quad \Delta P = 0.000{,}569 \frac{\mu^{0.25} s^{0.75} Q^{1.75} L}{D^{4.75}} \text{ .... (27)}$$

The foregoing equations can also be expressed in the following forms which utilize the more readily available kinematic viscosity values:

For laminar flow: $$f = 0.02025 \frac{\nu D}{Q} \qquad (28)$$

$$P = 0.000{,}273 \frac{\nu s Q L}{D^4} \qquad (29)$$

For turbulent flow: $$f = 0.042 \left(\frac{\nu D}{Q}\right)^{0.25} \qquad (30)$$

$$P = 0.000{,}569 \frac{\nu^{0.25} s Q^{1.75} L}{D^{4.75}} \qquad (31)$$

When calculating pressure losses for pressure lines, it is extremely important that the correct fluid viscosity values be used. The tabulation of kinematic viscosities at -50F, 0F, and +50F fluid temperatures in Table 11 shows that, for MIL-H-5606 fluid, the viscosity at 4,000 psi is approximately double that at atmospheric pressure; and, that the multiplication factor increases as fluid temperature decreases. At higher pressures, the multiplication factor is significantly higher.

TABLE 11 MIL-H-5606 FLUID VISCOSITIES AT ELEVATED PRESSURES

MIL-H-5606 Fluid Pressure (psi)	atmos.	2,000	4,000	6,000	8,000	10,000
Kinematic Viscosity at -50F (cs)	740	1,200	1,900	3,200	5,000	8,000
Kinematic Viscosity at 0F (cs)	100	140	200	285	410	580
Kinematic Viscosity at +50F (cs)	30	42	56	76	100	140

The foregoing values were taken from Figure 13 in AIR 1362 (Reference 2). Similar data for the candidate fluids would be required before accurate pressure-loss analyses can be made. However, if it is assumed that the pressure multiplication factors for the candidate fluids will be similar to those for MIL-H-5606 fluid systems, the following equations can be used to estimate the change in tubing diameters which would be required to convert an existing MIL-H-5606 fluid system for use with a candidate fluid:

For laminar flow: $$\frac{D_2}{D_1} = \left(\frac{\mu_2}{\mu_1}\right)^{0.25} \qquad (32)$$

$$\frac{D_2}{D_1} = \left(\frac{\nu_2 s_2}{\nu_1 s_1}\right)^{0.25} \qquad (33)$$

For turbulent flow: $$\frac{D_2}{D_1} = \left(\frac{s_2}{s_1}\right)^{3/19} \left(\frac{\mu_2}{\mu_1}\right)^{1/19} \qquad (34)$$

$$\frac{D_2}{D_1} = \left(\frac{s_2}{s_1}\right)^{4/19} \left(\frac{\nu_2}{\nu_1}\right)^{1/19} \qquad (35)$$

where

μ = absolute viscosity
ν = kinematic viscosity
ρ = fluid density
subscript 1 = reference fluid
subscript 2 = fluid of interest

These tubing size ratios were plotted against fluid design temperatures in Figures 16 and 17 to show the required increases in tubing sizes for both candidate fluids compared to tubing sized for an MIL-H-5606 fluid system. Figure 16 shows that tubing size increases for pressure and return lines designed for laminar flow conditions could be somewhat smaller for an E6.5 fluid system than for an AO-8 fluid system at all design temperatures above -40F. Figure 17 shows that the size increases for such lines designed for turbulent flow conditions could be smaller for an E6.5 fluid system at all design temperatures.

2.3.4.1.2 Velocity Limits and Pressure Peaks

As previously noted, MIL-H-5440 specifies that "peak pressure resulting from any phase of the system operation shall not exceed 135 percent of the main system, subsystem, or return system pressure when measured with electronic equipment or equivalent." The velocity of the hydraulic fluid flowing through system tubing directly affects the magnitude of peak pressure surges when an abrupt valve closure is initiated under a high-flow condition. The magnitude of the pressure rise above the normal system pressure can be calculated from the following formula:

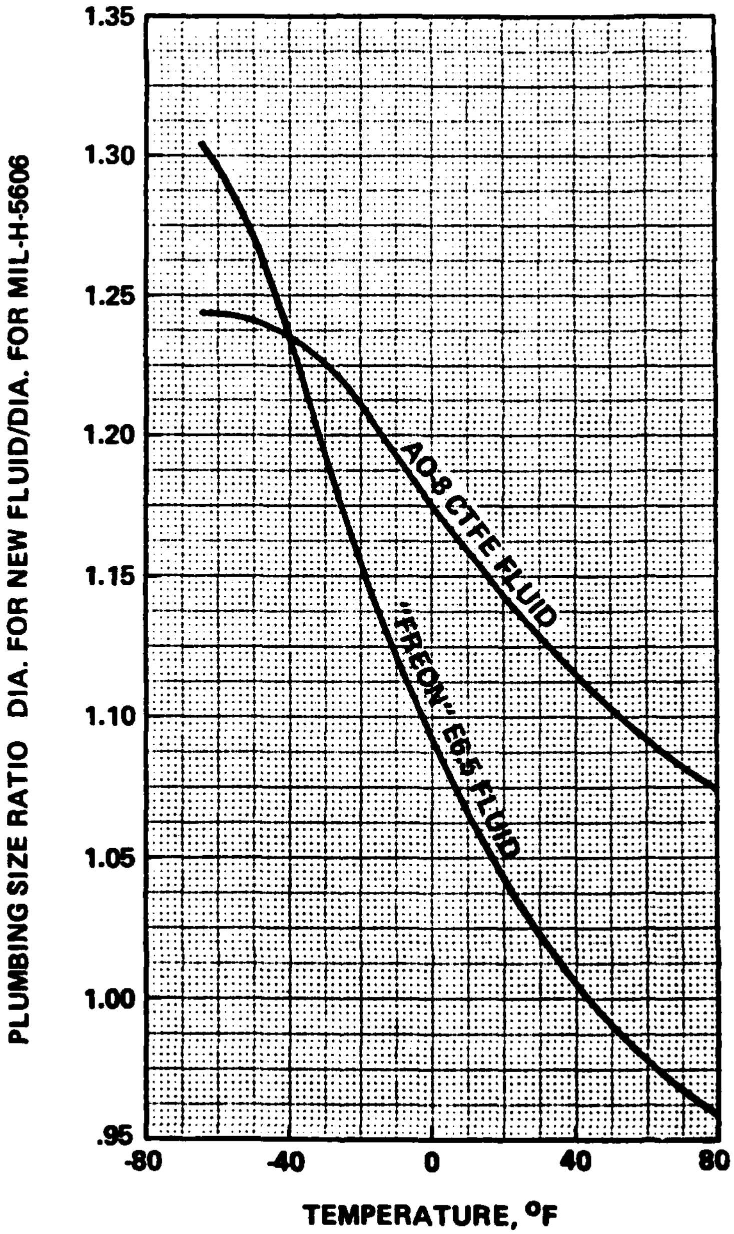

Figure 16. Laminar flow tubing diameter ratios for the two candidate fluids vs. MIL-H-5606 fluid

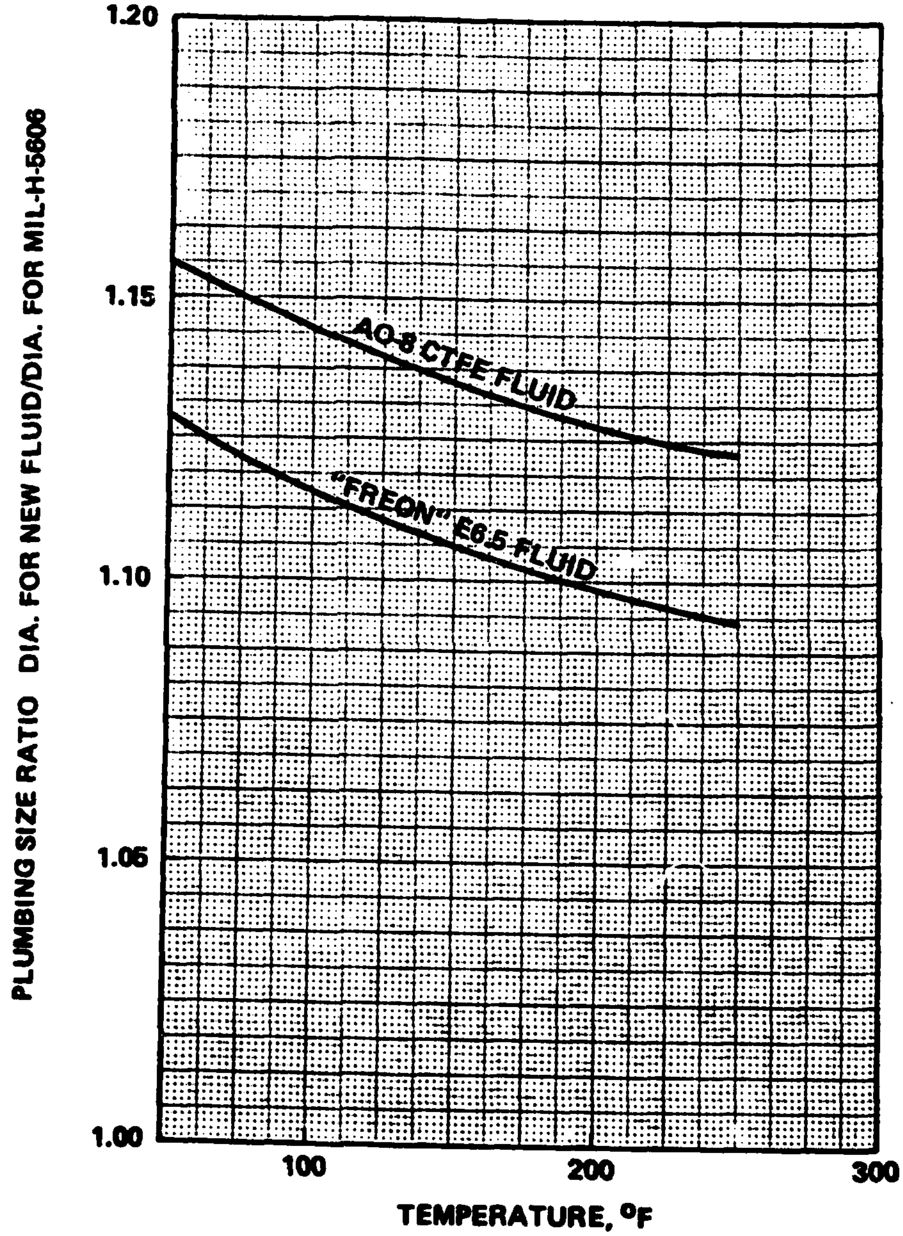

Figure 17. Turbulent flow tubing diameter ratios for the two candidate fluids vs. MIL-H-5606 fluid

$$\Delta P = 12V\sqrt{\beta\rho} \qquad (36)$$

where:

ΔP = Pressure rise, pounds per square inch (psi)
V = Original fluid velocity, feet per second (fps)
β = Fluid bulk modulus, pounds per square inch (psi)
ρ = Fluid mass density, $lb\text{-}sec^2/in^4$

As seen in Table 12, typical limiting velocities for MIL-H-5606 fluid, AO-8 CTFE fluid, and "Freon" E6.5 fluid in 3,000-psi systems, as required to keep the pressure rise within 35 percent of system pressure (1,050 psi), will be on the order of 25, 19, and 21 feet per second respectively depending upon the fluid operating temperature and the assumed fluid system compliance. It should be noted that typical values of fluid system compliance are somewhat smaller than the fluid bulk modulus. Laboratory measurements of fluid bulk modulus are generally always made with all entrained air and other gasses carefully removed, whereas in actual fluid systems the compliance is reduced by the effect of such entrained gasses and by the elasticity of the tubing and hoses.

TABLE 12 COMPARATIVE FLUID VELOCITY LIMITS

Parameter	MIL-H-5606 Fluid	AO-8 CTFE Fluid	"Freon" E6.5 Fluid
Density (at 77F) (g/cm^3)	0.84	1.86	1.82
Adiabatic-Tangent Bulk Modulus (77F @ 3,000 psi) (psi)	273,300	240,700	157,300
Assumed Fluid/System Compliance	150,000	125,000	100,000
Fluid Velocity for 1,050-psi Pressure Rise due to sudden Valve Closure (fps)	25	19	21

It may be noted that these values are greater than the historical fluid velocity limitation of 15 feet per second which was specified in the original issue of MIL-H-5440 and in all subsequent revisions through Revision D. However, in MIL-H-5440E and in subsequent issues, the reference to the 15 fps limitation was replaced with the following requirement:

"Fluid velocity limitations - Tubing size and maximum fluid velocity for each system shall be determined considering, but not limited to, the following:

(a) Allowable pressure drop at minimum required operating temperatures.

(b) Pressure surges caused by high fluid velocity and fast response valves.

(c) Back pressure in return lines, as it may affect brakes and pump case drain lines.

(d) Pump inlet pressure, as affected by long suction lines, and a high response rate variable pump. Consideration should be given to both pressure surges and cavitation."

2.3.4.2 <u>Pump Suction Lines</u>

The size of the pump suction lines must be adequate to ensure flow to the pump upon startup, and adequate pressure at the pump inlet port to preclude cavitation damage of the pump during all expected flow demands. To move hydraulic fluid from the system reservoir to the pump, sufficient reservoir pressure must be provided to both overcome the steady-state pressure losses in the pump suction line and also to accelerate the column of fluid in the suction line. Cavitation will occur if the system designer fails to match the flow response of the inlet system to the response required by the discharge flow demands from the pump.

The steady-state pressure losses can be calculated with the same formulae used for pressure and return lines making sure that the atmospheric-pressure viscosity values are used. The additional pressure required to ensure adequate response can be determined from the basic equation for acceleration force (F = ma) as follows:

$$F = \rho AL \left(\frac{dV}{dt}\right) = \rho AL \left(\frac{dQ/A}{dt}\right) = \rho L \left(\frac{dQ}{dt}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (37)$$

The equivalent flow-response pressure requirement can be expressed as follows:

$$P_{resp} = \frac{F}{A} = \frac{\rho L}{A}\left(\frac{dQ}{dt}\right) = \frac{\rho L}{\pi D^2/4}\left(\frac{dQ}{dt}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (38)$$

When converted to a form which utilizes the commonly used engineering units for fluid density (grams per cubic centimeter), for line length (feet), and for flow rate (gallons per minute), the foregoing formula appears as follows:

$$P_{resp} = 0.0055 \frac{\rho L}{D^2}\left(\frac{dQ}{dt}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (39)$$

where

P_{resp} = Flow-response pressure requirement (psi)

dQ/dt = Flow-response requirement (gpm/s)

L = Line length (ft)

D = Tube inside diameter (in)

The required pump suction line tubing diameter can be calculated by utilizing the following equation for the total suction line pressure requirement. However, it should be noted that at least two operating conditions need to be examined to ensure sufficient pressure at the pump inlet port to "prime" the pump at the minimum operating temperature (when the fluid viscosity is high), and to ensure that the flow through the suction line responds to pump output flow demands sufficiently to preclude damaging cavitation of the pump.

$$P_{resv} = P_{crit} + P_{resp} + \Delta P_{comp} + \Delta P_{line} \quad \text{.....................} \quad (40)$$

where

P_{resv} = Reservoir pressure

P_{crit} = Pump critical inlet pressure

P_{resp} = Flow-response pressure requirement

ΔP_{comp} = Pressure loss in suction-line components, ie: shutoff valve, check valve, self-sealing disconnect coupling, etc.

ΔP_{line} = Pressure loss in suction-line tubing and hose

The response component (P_{resp}) only becomes significant if a hydraulic pump with a long suction line is required to provide flow for fast-response high-flow actuator demands. For the two candidate fluids, this term is approximately equal; and, since density is the only fluid property involved, it would be more than twice that for MIL-H-5606 fluid.

The suction-line pressure-loss component would most likely be determined for laminar flow conditions since they are generally designed for the minimum pump full-flow design operating temperature. As noted in Eq. (32), the ratio of required tube diameters for two comparative fluids is a function of their absolute viscosities.

Two common suction line design points are full pump flow at an assumed bulk fluid temperature at takeoff (e.g. -20F for combat aircraft and +50F for transports) and small flow sufficient to permit system warmup from the minimum cold soak temperature (e.g. -65F). The suction line size requirement should be calculated at all design points with the largest diameter then used in the design.

At -65F, the absolute viscosities of the candidate fluids are nearly equal to each other, but considerably higher than for MIL-H-5606 fluid. These higher viscosities would require suction line diamaters 25 to 30% larger than for MIL-H-5606 fluid. At -20F, the absolute viscosities of the candidate fluids are roughly 50% higher than for MIL-H-5606 fluid thus requiring a 10% increase in size. At +50F, E6.5 and MIL-H-5606 fluids have essentially the same absolute viscosity, but AO-8 has a viscosity 25% higher which would require a suction line diameter increase of 5%.

Either of the above design points might dictate the suction line diameter but neither can be assumed to be the more likely one. Suction line

sizing is highly dependent on aircraft configuration (line length), as well as a function of mission related design criteria (flow and temperature) and component design features (reservoir and pump critical-inlet pressure). As these all vary widely, among the various type aircraft, no absolute comparison can be made between either of the fluids.

2.3.5 Impact on System Weight

The primary impact of the candidate fluids on the weight of hydraulic system components, as compared to components designed for use with fluid per MIL-H-5606, would be due to the contained fluid rather than the housings. However, significant increases in the weight of hydraulic tubing would also be expected with the use of the candidate fluids due to the increased sizes necessary to maintain pressure losses within desired limits. The increased line sizes would increase the volume of fluid in the system; and, this, in turn, could require larger reservoirs and possibly larger heat exchangers.

Balancing those weight increases to some extent, would be the weight decreases which could be realized by eliminating pump suction-line shutoff valves and other fire safety provisions such as firewalls and shrouds installed to isolate hydraulic fluid from ignition sources.

The largest increment of added weight in most systems will be in the pressure and return fluid distribution lines. Where larger sizes are required to maintain pressure losses within limits, they will also impact the weight of the tube fittings, clamps, line blocks, and possibly even some support brackets.

To get a better view of what the line diameter changes mean in weight terms, a weight ratio equation was developed for use with the diameter ratios determined through the use of Eq. (32) through Eq. (35).

Starting with the weight relation $W_{TOTAL} = W_{TUBE} + W_{FLUID}$, the following wet tube weight ratio was used for comparison of the candidate fluid system tubing weights with the MIL-H-5606-system tubing weight.

$$\frac{W_2}{W_1} = \frac{D_2^2}{D_1^2}\left(\frac{\rho_2 + 4\rho_t\left[\frac{t_2}{D_2} + \frac{t_2^2}{D_2^2}\right]}{\rho_1 + 4\rho_t\left[\frac{t_1}{D_1} + \frac{t_1^2}{D_1^2}\right]}\right) \quad \text{......................} \quad (41)$$

where

W = weight, wet tube
ρ_t = density, tube material
t = thickness, tube wall
Subscript 1 = reference fluid parameters (MIL-H-5606)
Subscript 2 = parameter for fluid of interest

Figure 18 is a plot of the pressure line weight ratio for the candidate fluids relative to MIL-H-5606 fluid over the laminar flow design temperature range. It shows that the fluid-filled (wet) weight of tubing designed for either of the candidate fluids at -40F would be approximately twice the wet weight of tubing for an MIL-H-5606 fluid system. It also shows that, at higher design temperatures, the weight penalties for both candidate fluids would be lower, and that the tubing system weight for the AO-8 fluid would be higher than for the E6.5 fluid.

Return-line weight ratios were determined similarily; but, in order to reduce the equation to a workable form, the tube wall thickness and tubing material density were incorporated. With aluminum return lines having a common wall thickness of .035 inch for all sizes (transport aircraft practice) and a density of .098 pounds per cubic inch, the equation becomes:

$$\frac{W_2}{W_1} = \frac{\rho_2(D_2/D_1)^2 D_1{}^2 + .01372(D_2/D_1)D_1 + .00049}{\rho_1 D_1{}^2 + .01372D_1 + .00049} \quad \ldots\ldots\ldots\ldots\ldots\ldots (42)$$

A visual inspection of this equation shows that the weight ratio not only is a function of temperature through the diameter ratios but also is a function of the reference line size (D_1). This results is a family of curves if plotted as the pressure lines are in Figure 18. Figure 19 shows the weight ratios for the two candidate fluids at temperatures of -20F (a typical full-flow temperature for combat aircraft) and +50F (a typical full-flow temperature for transport aircraft) plotted as a function of line size. It shows that the weight penalty for the AO-8 fluid is somewhat higher than for the E6.5 fluid at both temperatures, and that the weight penalties are larger for the larger tube sizes.

The next step was to take these theoretical weight ratios and apply them to an advanced airplane design. The most readily available detail hydraulic system data for an advanced aircraft design was that of the Air Force/Boeing YC-14 Advanced Medium STOL (AMST) airplane. The results of that weight study are presented in Table 13 which shows a significant increase in hydraulic system weight for both candidate fluids.

TABLE 13 ESTIMATED WEIGHT INCREASES TO REDESIGN THE YC-14 AMST PROTOTYPE AIRCRAFT HYDRAULIC SYSTEM FOR THE CANDIDATE FLUIDS

Fluid	Density g/cm^3 @ 77F	System Weight lb	Weight Increase lb	Weight Increase %
MIL-H-5606	0.84	7,202		
AO-8 CTFE	1.836	9,189	1,987	27.6
"Freon" E6.5	1.815	8,652	1,450	20.1
			537	7.5

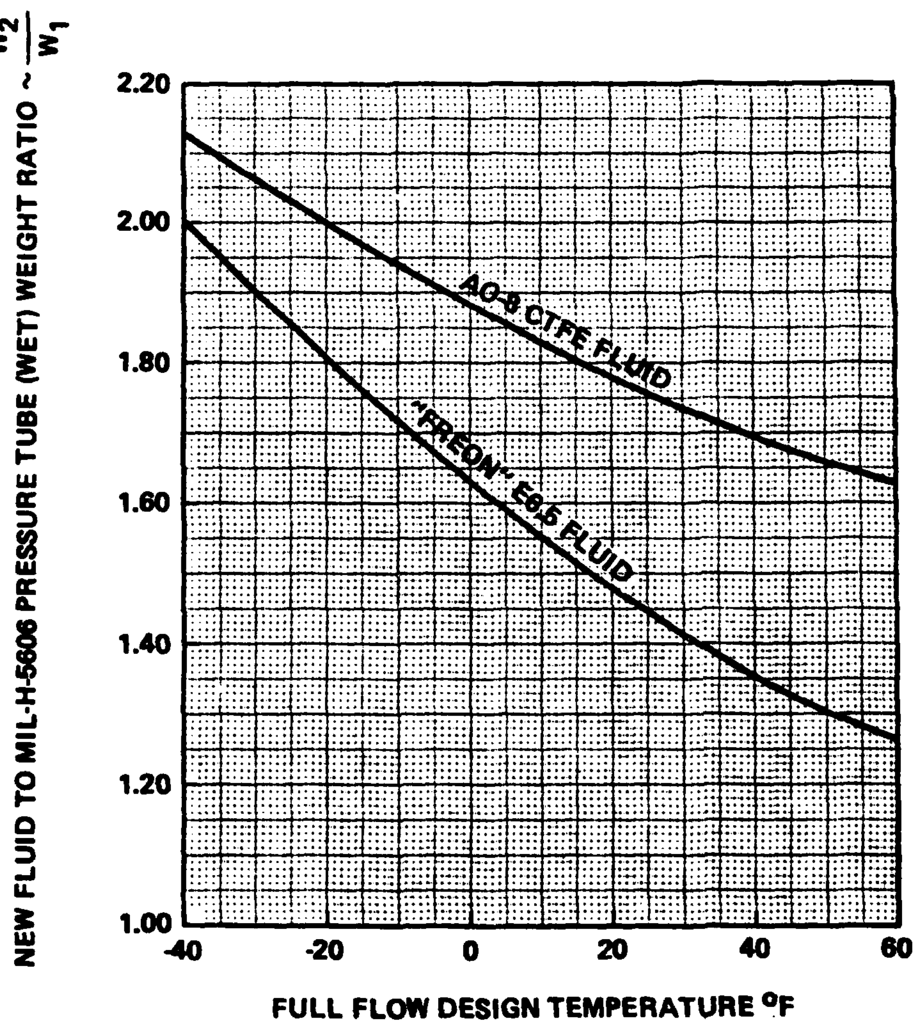

Figure 18. Laminar-flow pressure-tube weight ratio for the two candidate fluids vs. MIL-H-5606 fluid

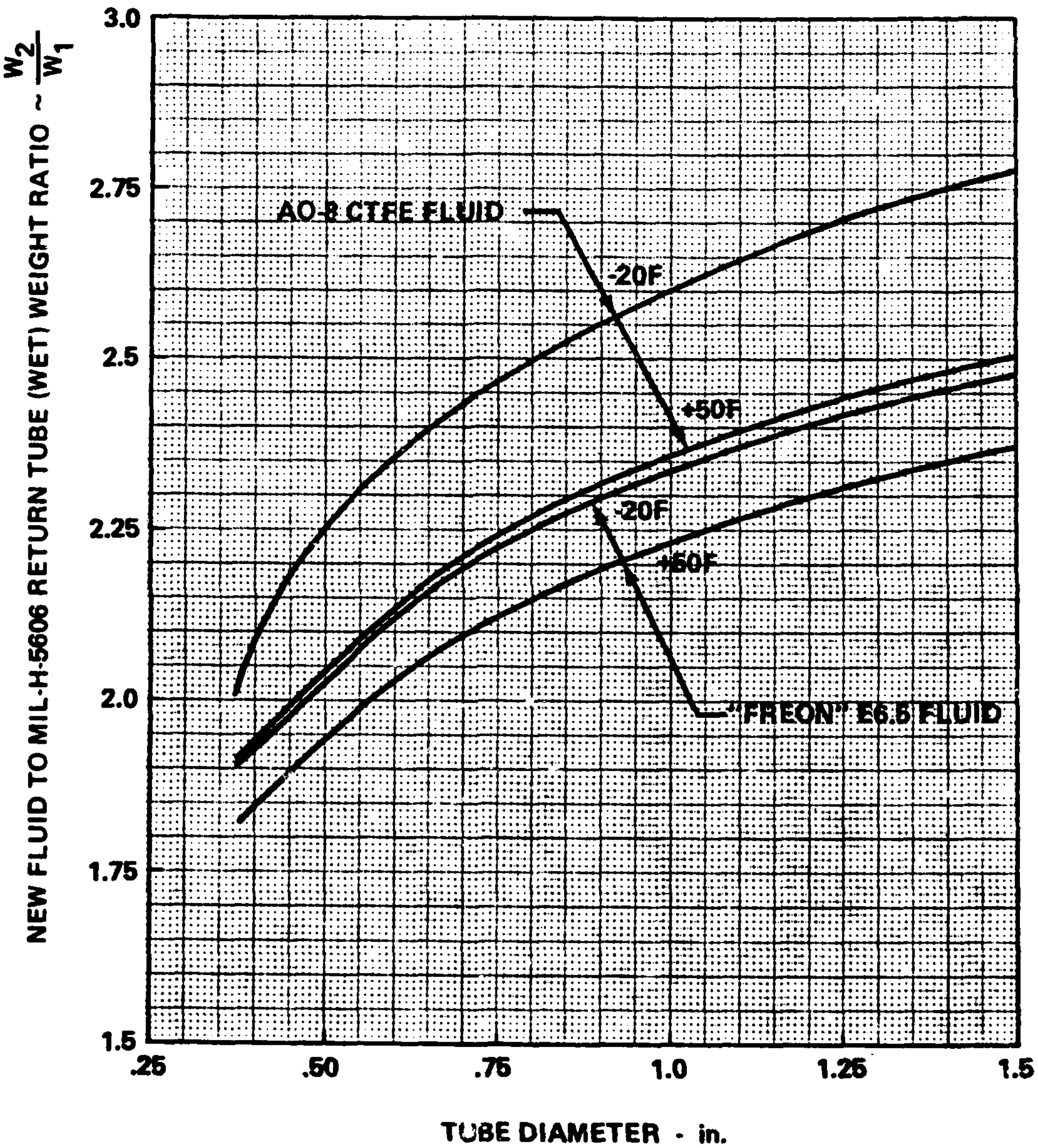

Figure 19. Return-line weight ratio for the two candidate fluids vs. MIL-H-5606 fluid

Redesigning for the AO-8 fluid would increase the weight of the overall hydraulic system by 1,987 lb (27.6%). Redesigning for the E6.5 fluid would increase the weight of the overall system by 1,450 lb (20.1%) which is 537 lb less penalty than the increase for the AO-8 fluid.

Similar estimates were made for the proposed C-14A production AMST aircraft. The estimated baseline weight was somewhat lighter than for the prototype aircraft due to expected design optimizations. However, as shown in Table 14, the estimated weight increases for the candidate fluids were nearly equal to those estimated for the prototype system.

TABLE 14 ESTIMATED WEIGHT INCREASES TO INCORPORATE THE CANDIDATE FLUIDS IN THE PROPOSED C-14A AMST HYDRAULIC SYSTEM

Fluid	Density g/cm^3 @ 77F	System Weight lb	Weight Increase lb	Weight Increase %
MIL-H-5606	0.84	6,347		
AO-8 CTFE	1.836	8,060	1,713	27
"Freon" E6.5	1.815	7,616	1,269	20

For smaller aircraft such as fighters, close support aircraft,and helicopters, where the distribution tubing runs are shorter, a smaller weight penalty would be expected.

2.3.6 Means to Reduce the Weight Penalties

A number of methods for reducing the weight penalties can be considered. One of these is to use lower viscosity versions of the candidate fluids in order to reduce tubing sizes and fluid volume. As shown in Appendix B, the weight penalty for hydraulic tubing runs, including the fluid contained therein plus the attachment clamps and end fittings, designed for use with the Halocarbon AO-8 CTFE fluid could be reduced some 57% with the smaller sizes allowed by the use of the lower-viscosity Halocarbon 1.8/100 CTFE fluid. However, this would require hydraulic pumps and motors which can operate without lubrication failure with the reduced viscosity at their maximum operating temperatures, ie: 0.94 centipoise at 275F compared with 2.6 cp for MIL-H-5606 fluid.

Other means to reduce the weight penalty is to reduce the fluid volume involved in fluid flow, such as through the use of higher system operating pressures or the use of load-adaptive actuation systems, in order to reduce tube sizes; or reducing tubing length through the use of integrated actuator packages or satellite hydraulic systems located at the remote points of usage around an aircraft. However, none of the latter techniques is unique to the candidate fluids. They could also be used to reduce the weight of a hydrocarbon-base hydraulic fluid system.

Another way to reduce the weight impact of a candidate fluid is to use it only in those portions of an overall system which are proximate to ignition sources such as engines and wheel brakes. One weight-effective approach is to use it only in the wheel brake systems. Air Force experience indicates that approximately two-thirds of their aircraft hydraulic fires occur in the wheel well areas due to fluid leaking onto hot wheel brake assemblies. Use of a nonflammable fluid only in the brake systems would provide a significant reduction in hydraulic fluid fires for only a relatively small weight penalty.

Tests of a two-fluid brake hydraulic system conducted on the Fireproof Brake Hydraulic System Research and Development Program, reported in Reference 6, indicated that the concept is feasible. In that program, in which a KC-135 aircraft brake system was tested in the laboratory, Halocarbon AO-2 CTFE fluid was used in the brakes and brake lines downstream of a modified KC-135 brake deboost valve and MIL-H-5606 fluid upstream. The basic operation and control characteristics of the brake system were not affected by the two-fluid configuration; and, the hardware modifications had virtually no effect on system performance. As noted in the final report for that program, the increased density of the CTFE fluid did affect the dynamic response of the brake hydraulic system which resulted in an indicated increase in aircraft stopping distance over that obtained with the original MIL-H-5606 fluid system. However, analysis indicated that the performance lost by changing to the CTFE fluid could be regained by increasing the hydraulic line sizes, by using hard tubing rather than hoses, or by retuning the antiskid control box.

2.3.7 Cost Impacts

To fully assess the cost impact of introducing and using a new hydraulic fluid, the cost impact on system components, spares, and support equipment must be considered in addition to the fluid itself. It is also possible to estimate the impact on life cycle costs of any given weapons system.

2.3.7.1 Fluid Cost

The fluid costs quoted herein are subject to the validity of the estimates provided by the prospective manufacturers. In the industry survey conducted in 1976 to identify candidate fluids for evaluation, current prices for 3 gallons, 10 gallons, and 50 gallons were requested plus estimates of the 1980-1985 production prices for the following annual quantities: 10,000, 100,00, and 1,000,000 gallons. The quotations received for the two candidate fluids and the subsequent revisions were as follows:

6. S. M. Warren and J. R. Kilner, Fireproof Brake Hydraulic System, AFWAL-TR-81-2080, Boeing Military Airplane Co., Seattle, WA, September 1981.

DuPont "Freon" E6.5 Fluid (Initial Quotation)

1976 price:	$2,025	for 45 lb. (3 gal.)	=	$675/gal.
	3,750	for 150lb. (10 gal.)	=	$375/gal.
	15,000	for 750lb. (50 gal.)	=	$300/gal.

Projected price:	For 10,000 gal/yr:	$50/lb.	=	$750/gal.
(1976 quote for	For 100,000 gal/yr:	$15/lb.	=	$225/gal.
1980-1985	For 1,000,000 gal/yr:	$ 5/lb.	=	$ 75/gal.
production)				

DuPont also noted that they formally stopped the manufacture of all "Freon" series compounds in 1975, and that they would consider resumption only if suitable incentives (including of yield of not less than 15% net return on investmment over a five-year period) were provided. In a subsequent letter, however, they stated that they had reestimated their costs and project a price of about $14 per pound ($210/gal.) for a one-million-gallons/year production rate. Additionally, they estimated that a production facility would cost some $230 million and that they would be interested in producing the fluid only if they receive a guaranteed return contract.

Halocarbon AO-8 Fluid (Initial Quotation)

1976 price:	For 3 gallons:	$232.50/gal.
	For 10 gallons:	202.50/gal.
	For 50 gallons:	202.50/gal.

Projected price:	For 10,000 gal/yr:	$160/gal.
(1976 quote for	For 100,000 gal/yr:	$120/gal.
1980-1985	For 1,000,000 gal/yr:	$120/gal.
production)		

During discussions in 1977, they stated that the estimated price could be reduced; and, in a letter on Oct. 5, 1977, they confirmed their verbal quote of $55 to $75 per gallon for one million gallons per year. They further stated that their existing facility is capable of producing in the thousands of gallons magnitude, and that a new site and facility with larger growth potential was started in the fall of 1977.

In contrast, at that time the price for the standard mineral hydraulic fluid per MIL-H-5606 was $3 per gallon, the price for the new synthetic hydrocarbon fluid per MIL-H-83282, which was adopted for Army helicopters and Navy aircraft, was $7 per gallon, and the price for the phosphate ester fluids used in commercial jet aircraft was $15 per gallon. These costs and the cost in typical package sizes are summarized in Table 15.

TABLE 15 COMPARATIVE PRODUCTION FLUID COSTS

Fluid	Container Size 1 Gal	5 Gal	55 Gal
MIL-H-5606	$ 3	$ 15	$ 165
MIL-H-83282	$ 7	$ 35	$ 385
Phosphate ester	$ 15	$ 75	$ 825
AO-8 CTFE	$ 75	$ 375	$ 4,125
"Freon" E6.5	$210	$1050	$11,550

2.3.7.2 Component Cost

The basic fabrication costs for new system components would not be expected to be much different than for conventional components. However, there will certainly be some additional non-recurring costs for development, design, purchase of new test equipment, and qualification testing. Additional recurring costs for new materials such as hydraulic fluid and elastomers, special standard components such as seals, and for acceptance testing, can also be expected to add to component costs. In an informal survey with a number of hydraulic equipment suppliers, an average cost impact of fifty percent cost addition was estimated.

2.3.7.3 Support Equipment Cost

In addition to the new or modified test benches which will be required by the equipment suppliers, new or modified benches will be required by all affected Air Force using commands and repair facilities. The 1978 costs for a hydraulic test bench ranged from $25,000 to $45,000 and for hydraulic ground carts from $25,000 to $50,000. The cost for modifying an existing test bench or ground cart, to drain and flush all components and tubing and replace all seals, was estimated to be $3,000.

2.3.7.4 Impact On Aircraft Life-Cycle Cost

A grasp of the effect of introducing and using each of the two candidate fluids is better obtained by reviewing the estimated life-cycle cost (LCC) impact on a complete aircraft weapon system. The following equation illustrates in simplified form the cost elements involved in a LCC study.

LCC = DEVMT + OTHINV + PROD + O&S

where

LCC	=	Life-Cycle Cost	
DEVMT	=	Development Costs	
OTHINV	=	Other Investment Costs	Non-recurring Costs
PROD	=	Production Cost	
O&S	=	Operation and Support Costs	Recurring Costs

The increase in LCC for an aircraft hydraulic system designed for one of the candidate nonflammable fluids rather than mineral oil per MIL-H-5606 includes the following.

a. Development cost increases for:

Component design.
Qualification test fluid.

b. Production cost increases due to:

Use of new materials including O-rings.
Increased cost of fluid for acceptance testing.

c. Other investment cost increases for:

New hydraulic test benches and ground carts.

d. Operations and support costs increases for:

New standard spare parts.
Increased cost of hydraulic fluid.
Increased use of aircraft fuel due to the increase in system weight.

In comparing the relative LCC impact of the two candidate fluids on a typical aircraft weapon system, it was assumed that the increases in development costs, production costs, and other investment costs would be nearly equal. The major differences would be in the operation and support costs.

To illustrate the magnitude of such differences, a rudimentary comparison of the operation and support costs was made of the proposed production version of the Air Force/Boeing YC-14 Advanced Medium STOL Transport (AMST) prototype. This comparison was based on the following ground rules which were used in LCC studies for the C-14A AMST proposal effort:

a. 277 operationally active aircraft.
b. 20 year operational life.
c. 1.8 hr/day peacetime utilization.
d. 4.5 hr/day wartime utilization.
e. 36.8 cents/gal. fuel cost.

In addition, it was estiamted that, on average, the hydraulic fluid in each aircraft would be completely replenished once per year. This estimate was based on the record of commercial jet transport aircraft operators several of which average two replenishments per year with some 3200 flight hours resulting from an average daily utilization of 8.75 hours.

The results of this analysis, presented in Table 16, show that the predicted O&S costs for the AO-8 and E6.5 fluids would exceed the baseline fluid O&S cost by $318,900 and $653,130 respectively. It also shows that the majority of this cost increase, 71% for the AO-8 fluid and 90% for the E6.5 fluid is due to the high cost of the fluid compared to the MIL-H-5606 fluid.

TABLE 16 ESTIMATED LIFE-CYCLE OPERATION AND SUPPORT COSTS OF THE CANDIDATE FLUIDS COMPARED TO MIL-H-5606 FLUID FOR A C-14A AMST HYDRAULIC SYSTEM WITH FUEL AT 37¢/GALLON

	MIL-H-5606 Fluid	AO-8 CTFE	"Freon" E6.5
System fluid volume, gal.	130	152	140
System weights, lbs	6347	8060	7,616
System weight change, lbs	--	1713	1,269
Additional fuel weight required (to maintain aircraft range capability)		240	178
OW change charged to system, lbs		1953	1,447
Mission radius, nautical mi. (with fixed takeoff weight)	400	365	370
Fuel burn cost increase due to OW change, $/airplane	--	98,700	73,130
Fluid useage cost, $/airplane	--	220,200	580,000
Operation & support cost increase, $/airplane	--	318,900	653,130

TABLE 17 ESTIMATED LIFE-CYCLE OPERATION AND SUPPORT COSTS OF THE CANDIDATE FLUIDS COMPARED TO MIL-H-5606 FLUID FOR A C-14A AMST HYDRAULIC SYSTEM WITH FUEL AT $1.00/GALLON

	MIL-H-5606 Fluid	AO-8 CTFE	"Freon" E6.5
Fuel burn cost increase due to OW change, $/airplane	--	268,300	198,800
Fluid useage cost, $/airplane	--	220,200	580,000
Operation & support cost increase, $/airplane	--	488,500	778,800

An update of this study, based on fuel costing $1.00 per gallon is shown in Table 17 where it is seen that this O&S costs for the AO-8 and E6.5 fluids would exceed the baseline (MIL-H-5606 fluid system) O&S cost by $488,500 and $778,800 respectively.

2.3.8 Reliability/Maintainability Impact

In examining the candidate fluids' properties for potentially adverse effect upon system reliability and maintainability, close attention was given to the following properties and characteristics:

a. Lubricity

Poor lubricity can lead to failure of bearings and other highly loaded surfaces in hydraulic pumps, motors, actuators, and other components. Results of the Shell Four-Ball lubricity tests indicated that both candidate fluids compare favorably with MIL-H-5606 fluid at the test temperature: 167F. As shown in Table 3, the wear scar diameters for both candidates were smaller than for MIL-H-5606 fluid. However, at higher temperatures, the low viscosity of E6.5 fluid (below 2.0 centipoise at temperatures above 250F) may cause hydrodynamic lubrication failure unless adequate bearing areas are provided.

b. Compatibility with system and airframe materials

System component failures can be caused by chemical attack and corrosion of metals, and weakening or breakdown of elastomeric seals, plastic parts, and potting compounds. Attack on airframe metals and other finishes can lead to additional maintenance and possible failure. In the thermal, oxidation, and hydrolytic stability tests, both candidate fluids appeared compatible with most typical hydraulic component metal alloys. Other than some oxidation of the bronze material in AO-8 fluid, and its minor attack on bare tool steels and 15-5PH stainless steel in the presence of moisture in that fluid, the weight loss/gain values of the metal specimens were lower than for MIL-H-5606 fluid. Most plastics, except for silicone products in contact with the AO-8 fluid, appear to be unaffected by the candidate fluids. Elastomeric seals are of concern, especially with the AO-8 fluid. However, as previously noted in the Sections 2.3.2.5 and 2.3.3.2, promising elastomeric compounds have been found and tested by the Materials Laboratory with each of the candidate fluids.

c. Chemical and environmental stability

Fluid breakdown products such as solid particles, gels, and sludge can plug system filters and even small fluid passages, nozzles, and orifices. Varnish-like deposits on close fitting valve spools can cause sticking or even binding which could lead to loss of aircraft control. Both candidate fluids appear to be very stable for operation in a Type II hydraulic system environment. There was no excessive acid formation or other

evidence of fluid breakdown (as indicated by DTA and chromatograph testing) in the presence of typical system materials in the thermal, oxidation, and hydrolytic stability tests. Other than for the AO-8 fluid darkening to a black coloration, and the E6.5 fluid turning a cloudy white upon cooling below 150F in the 800-hour valve stiction tests, there was no evidence of fluid degradation in any of the testing. The valve slide breakaway forces of 1.0 and 0.5 pounds for AO-8 and E6.5 fluids respectively occuring during the valve stiction test, gave sufficient indication that valve stiction would not be a problem.

d. Additive stability

The breakdown of anti-oxidant, anti-wear, foam reducing and/or viscosity improvement additives can lead to deposit formation, failure, foaming, or adverse viscosity for which the additives were designed to protect against. Other than for a viscosity index improver in the AO-8 fluid and an additive to reduce its vapor pressure (a stable copolymer of vinyledene fluoride and chlorotrifluoroethylene), no additives were used in the candidate fluids evaluated for comparison. The Materials Laboratory did attempt to mix various viscosity improvement additives with the E6.5 fluid to increase its high-temperature viscosity, but found none that were soluble.

e. Foaming tendency

Fluids which foam readily can introduce entrained vapor or air into a hydraulic sytem which can lead to a significant reduction in the effective fluid bulk modulus and/or loss of pump prime. Low bulk modulus can cause compressibility losses in pumps and reduction in actuator stiffness, response, and resistance to flutter. Loss of pump prime can cause loss of its ability to keep its system pressurized. As previously noted in Section 2.3.3.5, both of the candidate fluids passed the MIL-H-5606 foaming test.

f. Valve erosion tendency

As previously noted in the discussion regarding the impact of the candidate fluids on flow and pressure control valves, fluids with electrical conductivity values which fall in the erosion prevalent band shown in Figure 6 can cause excessive valve erosion damage. This leads to the requirement for frequent internal leakage checks and a high maintenance cost for removal, repair, and replacement of eroded valves. Fortunately, it was found that both candidate fluids were sufficiently removed from the erosion-prevalent band to indicate that no electrochemical valve erosion potential exists.

In summation, it appeared that, if further testing verified that adequate O-ring seal life and reliability could be obtained, that neither candidate fluid would present a major reliability or maintainability problem.

It must be recognized, however, that new ground carts and test benches would be required and adequate precautions taken to avoid mixing other fluids with the selected candidate and vice versa.

2.3.9 Safety Aspects

In assessing the potential impact of a new hydraulic fluid upon personnel and aircraft safety, the primary properties of interest are: its toxicity, its flammability, and its reactivity with system and airframe materials.

The toxicological properties of the candidate fluids and the MIL-H-5606 and MIL-H-83282 fluids, which were obtained from the fluid suppliers, are summarized in Table 18. The Boeing Hygiene Group examined the data and summarized that "all of the hydraulic fluids listed show a low order of toxicity in animals, and present a low inhalation hazard to man due to low vapor pressure characteristics at ambient temperatures. None of the fluids are classified as primary eye/skin irritants. The toxicity data indicates that the fluoro, chloro/fluoro carbons are less irritating due to high chemical stability. Generally, the differences in toxicological properties are insignificant, considering the manner in which these fluids are used in industrial operations. (Refer to the attached table for specific toxicity data.) Safe handling procedures are similar for all listed fluids, i.e., avoidance of repeated/prolonged skin contact, and exposure to mists or vapors generated from heated fluids. The halogenated fluids present a greater hazard in event of thermal decomposition due to formation of highly toxic and corrosive pyrolysis products." It is noted here that in the Aminco Bomb/differential thermal analysis (DTA) tests, no oxidation temperature was found for either of the candidate fluids at temperatures up to the maximum test temperature of 900 to 1000F.

As discussed in Section 2.3.2.1, and as shown in Table 3, both candidate fluids have a high degree of nonflammability. In addition, the thermal, oxidation, and hydrolytic stability tests indicated that they both are relatively inert. However, due to a reported propensity of the Halocarbon AO-8 fluid to react spontaneously in aluminum screw threads, a series of explosive reactivity tests on both candidate fluids was run by the Phoenix Chemical Laboratory for the Materials Laboratory. As shown in the summary in Table 19, the AO-8 fluid showed only a minor reaction and the E6.5 fluid none at all.

Therefore, in view of their low order of toxicity, high degree of nonflammability, relative inertness, and low order of explosive reactivity, it was concluded that either candidate fluid would be safe to use in an aircraft hydraulic system.

2.4 SELECTION OF ONE FLUID FOR FURTHER EVALUATION

2.4.1 Summary of Fluid Property and Design Impacts Comparison

A detailed comparison of the significant properties of the two candidate fluids with the stated target values and with each other, as shown in Table 3, and their major impact on hydraulic components and overall systems were summarized as follows:

TABLE 18 HYDRAULIC FLUID TOXICITY DATA

PHYSIOLOGICAL PROPERTY	MIL-H-5606	MIL-H-83282	"FREON" E6.5	AO-8 CTFE
Acute Oral Toxicity	LD_{50} ⑥ >5 gms/kg ② None died at 5 gms/kg ⑦	LD_{50} 1.05 gms/kg ②	ALD ③ 25 gms/kg ② for E1, 2, 3, 7	LD_{50} >10 gms/kg ② Hooker Co. data
Acute Dermal Toxicity	LD_{50} >2 gms/kg ① None died at 2 gms/kg	MLD >5.0 gms/kg ①	ALD >37.5 gms/kg ① for E1, 2 >11 gms/kg for E7	Not determined by vendor. Expected to be of low toxicity based on tests conducted with similar materials.
Dermal Effects	Moderate skin irritant ① Not classified as a primary skin irritant. Index - 4.54	Moderate skin irritant ① Primary irritation Index - 3.8 on scale of 8	No local erythema ① (14 day exposure)	Not determined by vendor. Not expected to be a primary skin irritant based on tests conduced with similar materials.
Eye Irritation	Moderate eye irritant ① Scores of 3.7, 3.0, 0.2 at 24, 48, 72 hours, respectively. Not classified as a primary eye irritant.	Moderate eye irritant ① Average score of 24, 48 and 72 hour reading was 6.1 on a scale of 100.	Not determined by vendor. Not expected to be a primary eye irritant based on tests conducted with similar materials.	Not determined by vendor. Not expected to be a primary eye irritant based on tests conducted with similar materials.
Vapor Inhalation	Not expected to be an inhalation problem due to low vapor pressure at ambient temperatures.	Not expected to be an inhalation problem due to low vapor pressure at ambient temperatures.	ALC ⑤ >37% Vol. (4 hrs. test for E-11; >4.45% Vol. E-3 (saturated vapor plus mist) Higher molecular weight homologs have low vapor pressure, e.g., >0.01 mm/Hg at 77°F for E6.5.	Not expected to be an inhalation problem due to low vapor pressure at ambient temperatures.
Demyelination Effects.	Not determined by vendor.	Not determined by vendor.	Not determined by vendor.	Not determined by vendor.

① Experimental animal-Rabbit ② Experimental animal-Rat ③ ALD-Approx. Lethal Dose ④ MLD-Minimum Lethal Dose

⑤ ALC-Approx. Lethal Concentration ⑥ LD_{50}-Median Lethal Dose ⑦ gms/kg-Grams of materials per kilogram of body weight

TABLE 19 EXPLOSIVE REACTIVITY OF THE CANDIDATE FLUIDS WITH ALUMINUM
(PER ASTM METHOD D3115)

FLUID	RUN	SPARKS	BANGS	SMOKE	RESIDUE	REACTION TIME
E6.5	1	None	None	None	Darkened	No reaction
E6.5	2	None	None	None	Darkened	No reaction
E6.5	3	None	None	None	Darkened	No reaction
E6.5	4	None	None	None	Darkened	No reaction
E6.5	5	None	None	None	Darkened	No reaction
E6.5	6	None	None	None	Darkened	No reaction
AO-8	1	None	1(faint)	None	Darkened	40
AO-8	2	None	3(2 weak)	White	Darkened, soft semi-solid	5
AO-8	3	None	4(2 weak)	White	Darkened, soft semi-solid	3
AO-8	4	None	2(1 weak)	None	Darkened	3
AO-8	5	None	1(faint)	None	Darkened	3
AO-8	6	None	1(faint)	None	Darkened	12

a. They both meet the stated flammability requirements except for the following minor deficiencies:

   Their autoignition temperatures are somewhat below the 1300F target value. However, neither candidate fluid showed any tendancy to burn during the hot manifold ignition tests or atomized fluid flammability tests other than a slight flickering of each fluid on the back side of the hot manifold where the temperature may have exceeded the 1700F target value.

b. The density of both fluids is considerably greater than that of mineral fluid per MIL-H-5606, and both will increase system weight. The density of E6.5 fluid is slightly less than AO-8; and, although it would increase the weight of a production AMST hydraulic system by some 1269 lb., its weight penalty would be some 444 lb. less than for the AO-8 fluid.

c. The viscosity of both fluids at the -65F system cold soak temperature is greater than that of MIL-H-5606 fluid (2050 cp) and both are close to the target limit value of 5500 cp. The E6.5 fluid slightly exceeds the limit but neither are considered too viscous for pump cold starts. Fluids with even higher viscosity can be pumped; but, the time required to attain outlet pressure and flow increases with viscosity.

d. The viscosity of both fluids at the 300F maximum system temperature specified for these fluids are within the target limit vlaue of 1.0 cp. The viscosity of the AO-8 fluid (2.1 cp) is slightly above that of MIL-H-5606 fluid which is considered marginal at that temperature. The viscosity of the E6.5 fluid is less than 2.0 cp at all temperatures above 250F and less than that of the AO-8 fluid at all temperatures above -40F. This would result in smaller tube sizes than would be required for an AO-8 fluid system, and contributes to the lower weight penalty noted above. However, the low high-temperature viscosity of the E6.5 fluid was of concern for highly loaded surfaces dependent on hydrodynamic lubrication. Elements such as pump and motor bearings and actuator rod glands may require oversized areas and/or especially hardened materials, such as tungsten carbide, to prevent lubrication failure.

e. Both fluids have acceptable pour points and low temperature stability.

f. Both fluids have acceptable thermal, oxidation, and hydrolytic stability.

g. Both fluids have acceptable lubricity, as measured by the Shell Four-Ball wear scars; but, there was some concern about the hydrodynamic lubricity of the E6.5 fluid at high temperature.

h. The bulk modulus of the AO-8 fluid is nearly equal to the MIL-H-5606 fluid and well above the target value. The modulus for E6.5 fluid is below the target value, and could require that some flight control servoactuators be designed oversize to meet flutter stiffness requirements. The computer simulations of the B-52G/H elevator system indicated that its frequency response requirements could be met with either fluid although just barely with the E6.5 fluid. However, this won't always be the case. For other aircraft, e.g. the F-111, flutter stiffness rather than load requirements dictated the size of some servoactuators even with MIL-H-5606 fluid. With E6.5 fluid, the penalty would be greater.

i. The vapor pressure and foaming characteristics of both fluids indicated that they would have no penalizing effects on the design of reservoirs or other components.

j. The valve stiction tests indicated that neither fluid would deteriorate sufficiently to form deposits which could cause unacceptable valve breakout forces.

k. Elastomer compatibility tests indicated that there are several elastomer compounds which have the lab test properties required of a good seal. It was reported by the Materials Laboratory that ethylene propylene rubber had the best static results with E6.5 fluid. EPR O-rings are used very successfully (with phosphate ester fluids) in thousands of commercial transport aircraft. Little was known about the materials being evaluated for the AO-8 fluid, however. Although promising test results have been obtained with at least two materials, chlorinated polyethylene and a fluorinated phosphonitrilic compound, considerable testing under realistic system conditions would be required to validate that they are satisfactory for actual use.

l. Information from the two candidate fluid suppliers indicate that several electrical insulations and potting compounds capable of 300F environments are available for either candidate.

m. The electrical conductivity of each fluid was found to be sufficiently removed from the erosion-prevalent band to indicate that no electrochemical valve erosion potential exists.

n. The specific heat values and thermal conductivity of the two candidate fluids are essentially equal. The specific heats are lower than that of MIL-H-5606 fluid on a weight basis as is normally quoted; but, on a volume basis, are nearly the same indicating that there would be no reduction is system heat capacity. Their thermal conductivities are approximately half that of MIL-H-5606 fluid which would double the size of heat exchangers to obtain equal effectiveness.

In addition to the foregoing comparisons, it was noted that the AO-8 fluid has a potential advantage in that it can accept some additives whereas, as yet, no additives soluble in E6.5 fluid had been found. This could be a very important factor later on if an additive was found necessary to correct a fluid deficiency which may introduce undue component or system design or operational penalties or even make the fluid impractical to use.

The "Freon" E fluids have one similar advantage in that, although they appear incapable of accepting additives, the various homolog compounds could be mixed in different proportions in order to improve a deficiency in one or more properties. However, this may be at the expense of other properties.

2.4.2 Comparison of Price and Availability

The nonflammability requirements specified by the Aero Propulsion Laboratory and the Aeronautical Systems Division severely limited the number of potentially acceptable candidate fluids, and, the basic chemistry for a

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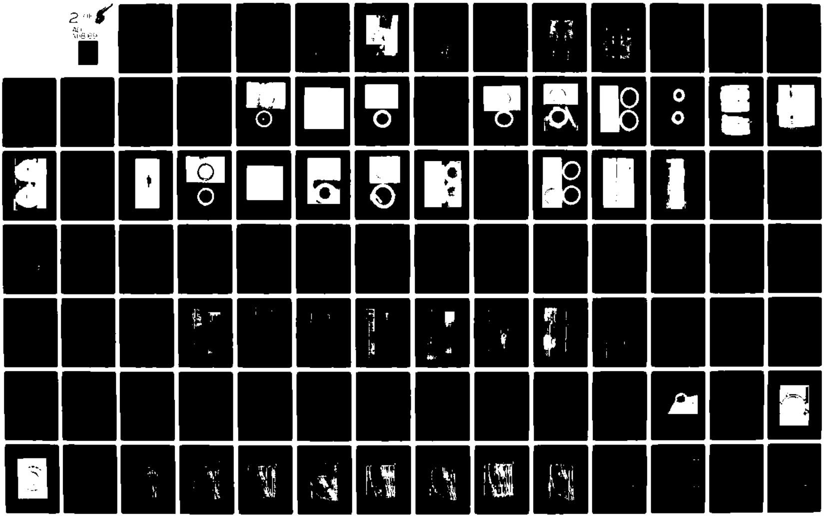


number of these candidates requires expensive synthesizing with low yield. As a result, these fluids will be considerably more expensive than fluids being used by the military at the present time. In addition, the unit cost of these fluids may not decrease very much with large quantity batching because the synthesizing would be no less expensive to accomplish with large quantities.

In narrowing the list of 20 proposed fluids to four and then to the two final candidates, the predicted price was a major factor in the selection. As shown in Table 2, vapor pressure, viscosity, and price were the deciding factors in selecting the E6.5 and AO-8 fluids. The two selected candidates had the best combination of physical properties and price; but, as shown in Table 19, they were both considerably more expensive than other commonly used hydraulic fluids.

In Table 21, the impact of both fluids on the operating and support life-cycle costs for a C-14A advanced medium STOL transport is shown. It is seen that both fluids would have considerable impact, and that the cost increase for the E6.5 fluid, would be twice that for the AO-8 fluid. Although, the predicted price for the E6.5 fluid was approximately three times that for the AO-8 fluid, its lower weight impact reduced the O&S life-cycle cost impact to a factor of two.

Of additional concern, was DuPont's positon regarding construction of a new production facility to manufacture the E6.5 fluid. As noted in their letter of January 6, 1978, they estimated an investment cost of $230 million (1978 dollars) -20% to +40%; and, they verbally stated that they would require a goverment guarantee of 15% net return on investment.

There was also concern regarding the availability of sufficient E6.5 fluid for hydraulic component testing. All of the available E6.5 stock was purchased for the screening tests. However, DuPont stated that other homologs of the "Freon" E fluid type were available; and, the Materials Laboratory stated that the E6.5 fluid could be duplicated accurately enough for a valid test program.

On the other hand, there appeared to be no problem in obtaining the AO-8 fluid. Halocarbon could provide fluid for additional testing in their existing facility, which is capable of producing in the thousands of gallons magnitude, and had started construction fo a new facility with larger growth potential. They also stated that their fluid is not proprietary and could be produced by other companies such as 3M, Allied Chemical, and Hooker Chemical.

2.4.3 The Recommendation For The Halocarbon AO-8 Fluid

As noted in the foregoing comparisons, the properties of the two candidate fluids are essentially equal in most respects. Their primary differences are in their viscosity, bulk modulus, and elastomer compatibility. Due to its lower viscosity in the normal operating temperature range, the E6.5 fluid would have a lesser impact on system weight than the AO-8 fluid. However, the low high-temperature viscosity of the E6.5 fluid may require heavier components to avoid hydrodynamic lubrication failure. Also, the low bulk modulus of the E6.5 fluid may require larger and heavier servoactuators to obtain adequate flutter stiffness.

Of equal concern, was the comparative elastomer compatibility of the two fluids. The E6.5 fluid appears to be compatible with ethylene propylene rubber which is a well proven O-ring material. However, the Materials Laboratory indicated that there had been a problem finding an O-ring elastomer which is compatible with the AO-8 fluid. Although two materials appeared promising, considerable testing would be required to validate them for actual use.

Of additional concern, was the relative cost and availability of the two fluids. This was considered the deciding factor. The price and investment cost for the E6.5 fluid was considered unacceptable. Therefore, the AO-8 fluid was recommended for component compatibility testing. It was further recommended that the Materials Laboratory continue its evaluation of elastomeric seal compounds and support Boeing in an effort to obtain a satisfactory elastomeric O-ring seal material.

3.0 COMPONENT COMPATIBILITY TESTS

Following the selection of the Halocarbon AO-8 fluid for further evaluation, the following components were subjected to compatibility tests by Boeing under typical aircraft hydraulic system operating conditions:

a. Static and dynamic elastomeric seals.
b. Hydraulic pumps.
c. Flight control servoactuator.

3.1 HYDRAULIC SEAL TESTS

3.1.1 Summary Of Prior Seal Testing At The Materials Laboratory

As noted in Section 2.3.2.5, the comparison of the two finalist fluid candidates included testing with various candidate elastomer materials by the Materials Laboratory. In the elastomer physical properties tests, which are documented in Reference 4, a number of seal materials were evaluated and the change in properties measured following immersion in the candidate fluids for 72 hours at 275F.

Based upon the results described in reference 4, O-ring seals based on Firestone's PNF elastomer compound R-205086 were recommended by the Materials Laboratory for further evaluation in the Boeing dynamic seal test system. It was recognized that the PNF material still had some shortcomings which might affect its sealing perfomance; but, it appeared to be the best candidate for successful operation as an O-ring seal. This was the deciding factor. The component design flexibility allowed by the use of O-ring seals provide decided overall economic advantages over the design compromises which must be made to accommodate other seal designs.

3.1.2 Boeing Seal Test Facility

The dynamic seal test system shown in Figures 20 and 21 is basically the same facility used by the Boeing Commercial Airplane Company for compatibility and qualification testing of phosphate ester fluids and ethylene propylene seals for commercial jet aircraft since 1963. It includes a hydraulic power supply, a power transfer assembly and a seal test cylinder. The hydraulic bench utilizes phosphate ester fluid; therefore, a fluid power transfer device was added to transmit pressure to the test fluid and test cylinder for this program.

3.1.2.1 Test Cylinder

The seal test cylinder, shown in Figure 22, has a one-inch diameter mechanically-stroked chrome-plated rod against which the dynamic seals were tested. The rod stroking apparatus has the capability to vary the stroke to four inches and the stroking rate between 25 and 400 cycles per minute. A cam-operated switch allows the pressure cycling to synchronize with the stroking. The cylinder has porting to allow pressuriztion by the hydraulic power supply and the reservoir, and to bleed air from the system.

The cylinder end caps act as holders for the test seals and rod bearing. One end of the cylinder contained standard O-ring seals with a

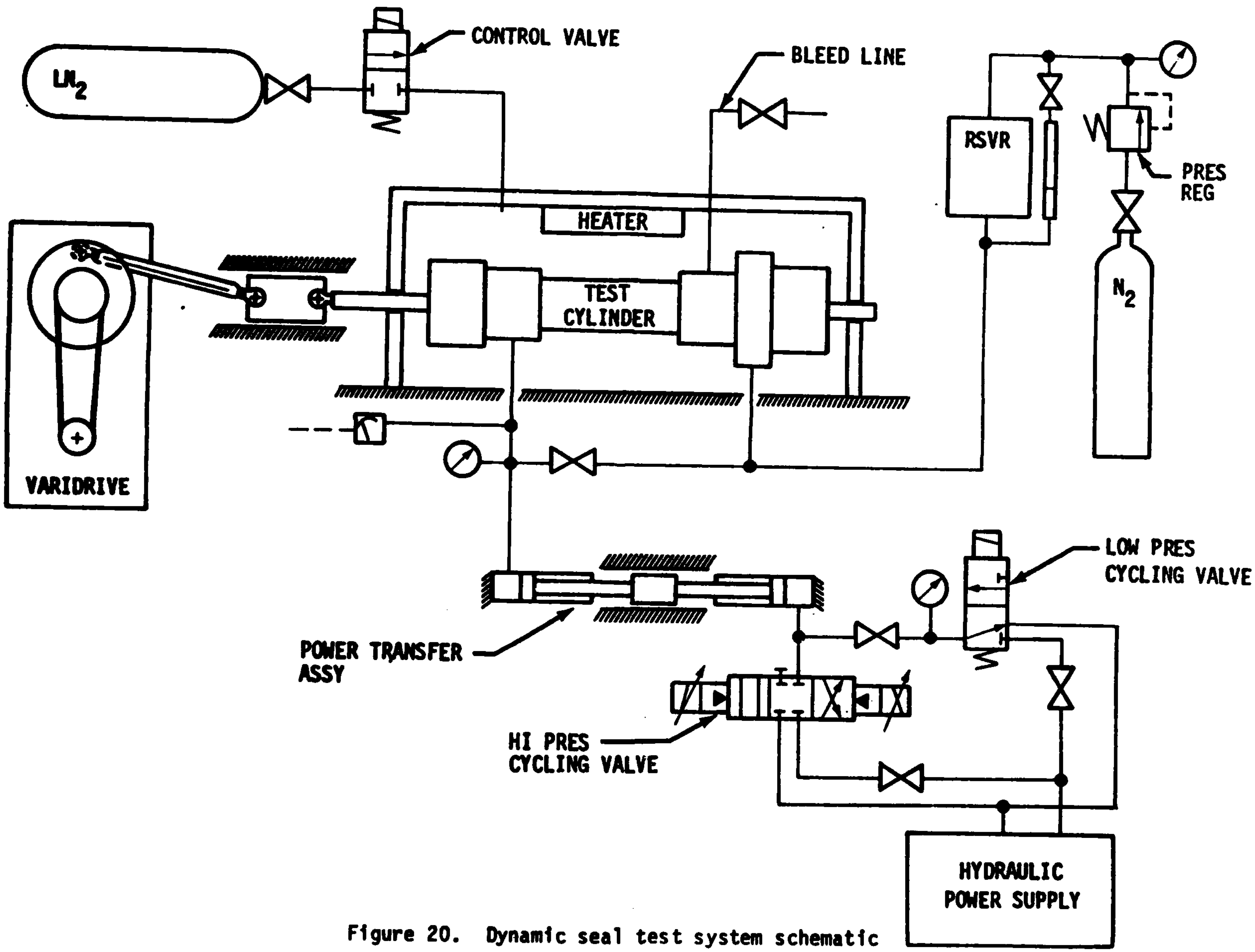


Figure 20. Dynamic seal test system schematic

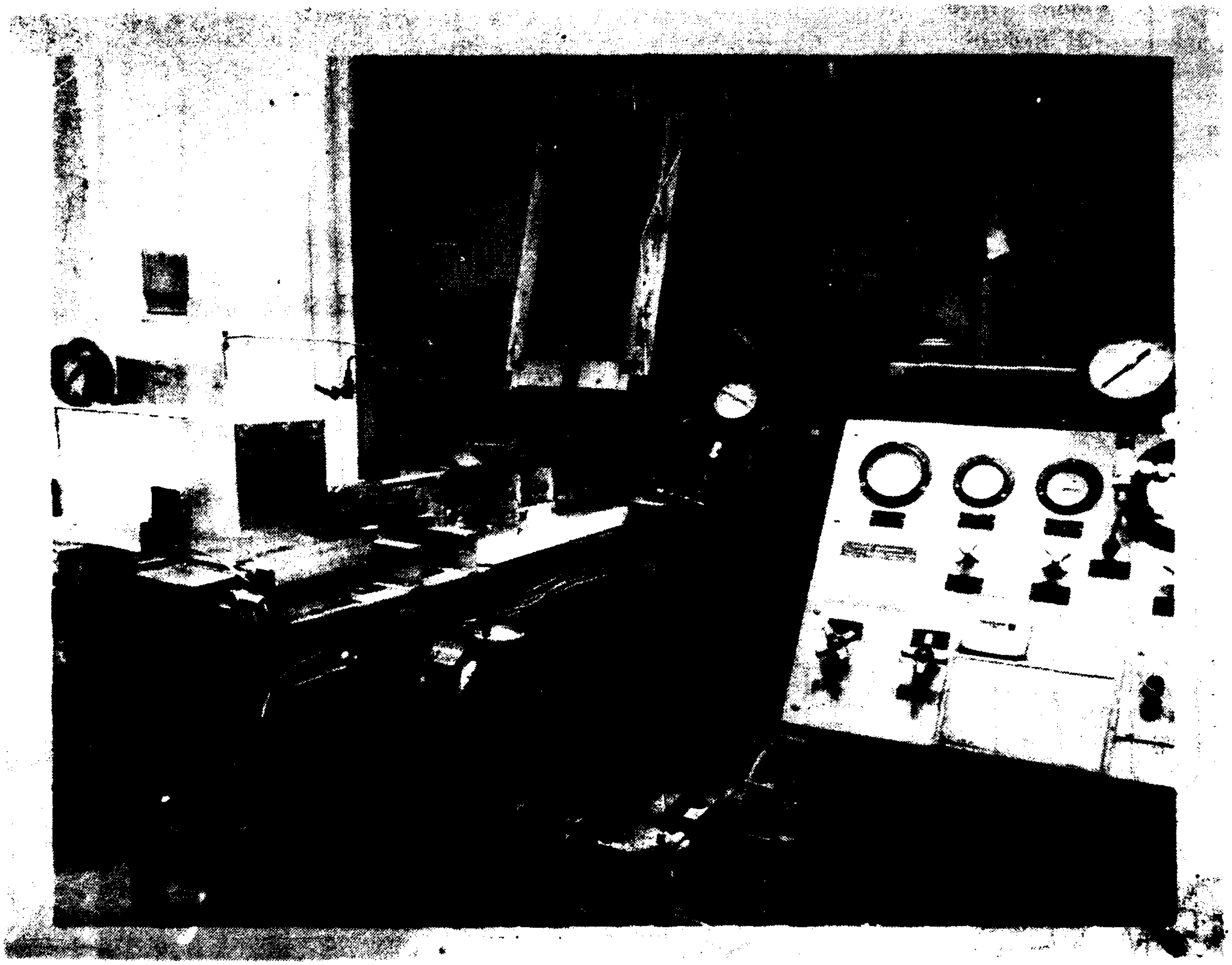


Figure 21. Dynamic seal test stand

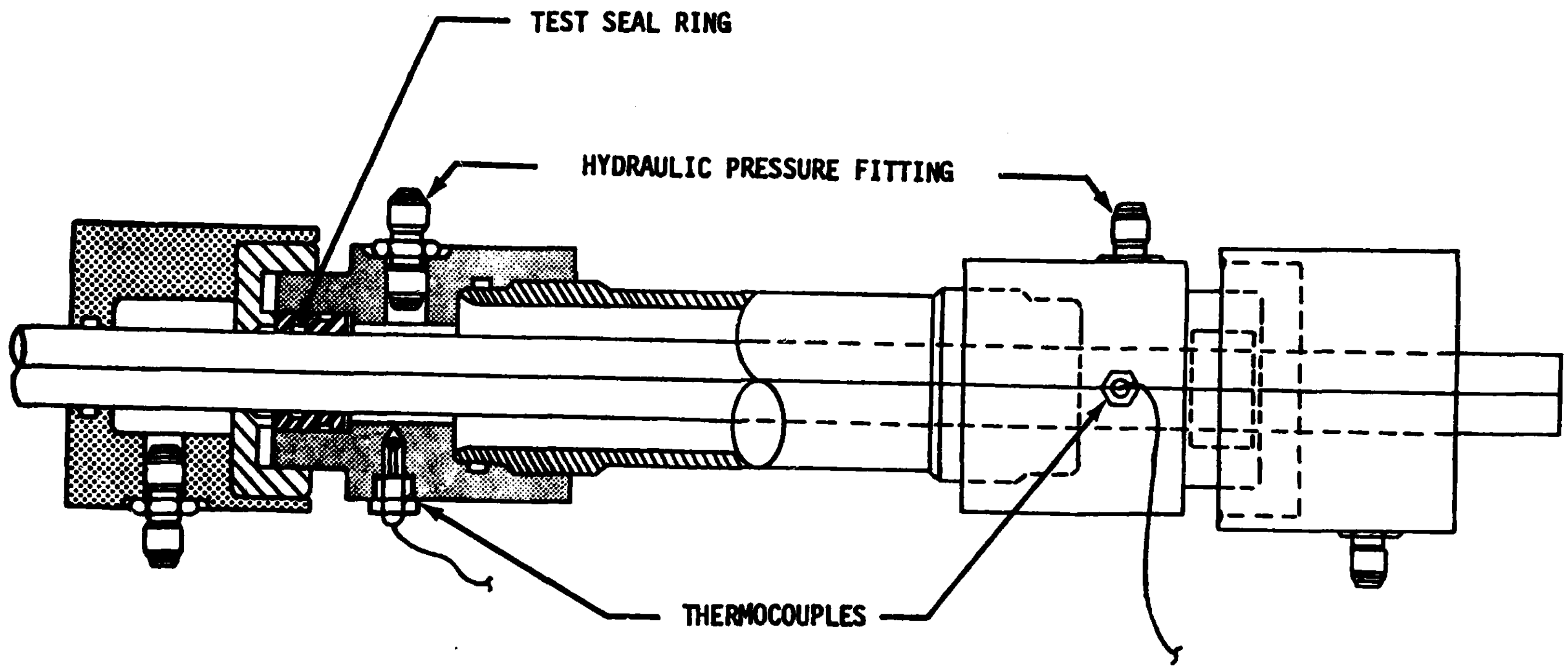


Figure 22. Seal test cylinder

single backup (anti-extrusion) ring in both static and dynamic applications as shown schematically in Figure 23. In the other end of the cylinder, (shown in Figure 24) a dynamic sealing system similar to the B-52 elevator servoactuator rod seals was used. It had a first-stage (high-pressure) multi-turn TFE ring and a second-stage (low-pressure) Foot Seal with reservoir pressure between the two. On the rod bearing, two external static seals with backup rings were used. The seal test cylinder also contained two larger diameter internal static O-rings with single backup rings. The remainder of the seals in the cylinder assembly were utilized to separate each test seal's leakage and guide it to a volume measuring vessel.

3.1.2.2 Instrumentation

The test facility included instrumentation to measure pressures, temperatures and number of stroking cycles. Pressure in the test cylinder was measured by gages and a transducer which provides a signal to an oscillograph recorder. Temperature pickups (thermocouples) were read at a digital display and recorded in a log. The thermocouples were located to measure the ambient temperature in the insulated box surrounding the test cylinder, and the temperature of the test seals as shown in Figure 22.

3.1.3 Seal Materials and Configurations

Two series of tests were run. One with MS28775 Buna N nitrile O-rings in MIL-H-5606 fluid, and the other with PNF O-rings in Halocarbon AO-8 fluid. The PNF O-rings were molded from Firestone compound, by Nichols Engineering, Inc. in Shelton, Connecticut, which has since been purchased by the Aerospace Products Division of the Lord Corporation. Standard O-ring dimensions per SAE Aeronautical standard AS 568 (Reference 7) were used.

The seal configurations tested were selected to determine the acceptability of military-standard O-ring rod seal and static seal designs, and of the Boeing B-52 elevator actuator two-stage rod seal design.

3.1.3.1 Military-Standard Rod Seal

The test cylinder contained a standard internal dynamic seal configuration, retained in a single-backup-ring-width groove per MIL-G-5514, as shown in Figure 23. For the test with MIL-H-5606 fluid, an MS28775-214 (1.0-inch I.D.) O-ring with a dual-turn MS28782-19 TFE backup (anti-extrusion) ring was used. For the AO-8 fluid test, the PNF O-ring was an AS568-214 size and the backup ring was an MS28782-19.

3.1.3.2 B-52 Two-Stage Rod Seal

A two-stage rod seal such as used in the B-52 elevator servoactuator, as shown in Figure 24, was used to assure the acceptability of this non-standard design prior to the subsequent servoactuator testing. For both fluids, the first-stage (high-pressure) seal was a W.S. Shamban Company P/N S12003-120-3 TFE triple-turn spiral ring retained in an internal groove of

7. SAE AS 568A, Aeronautical Standard, Uniform Dash Number System for O-Rings, Society of Automotive Engineers, Inc., Warrendale, PA, July 1974.

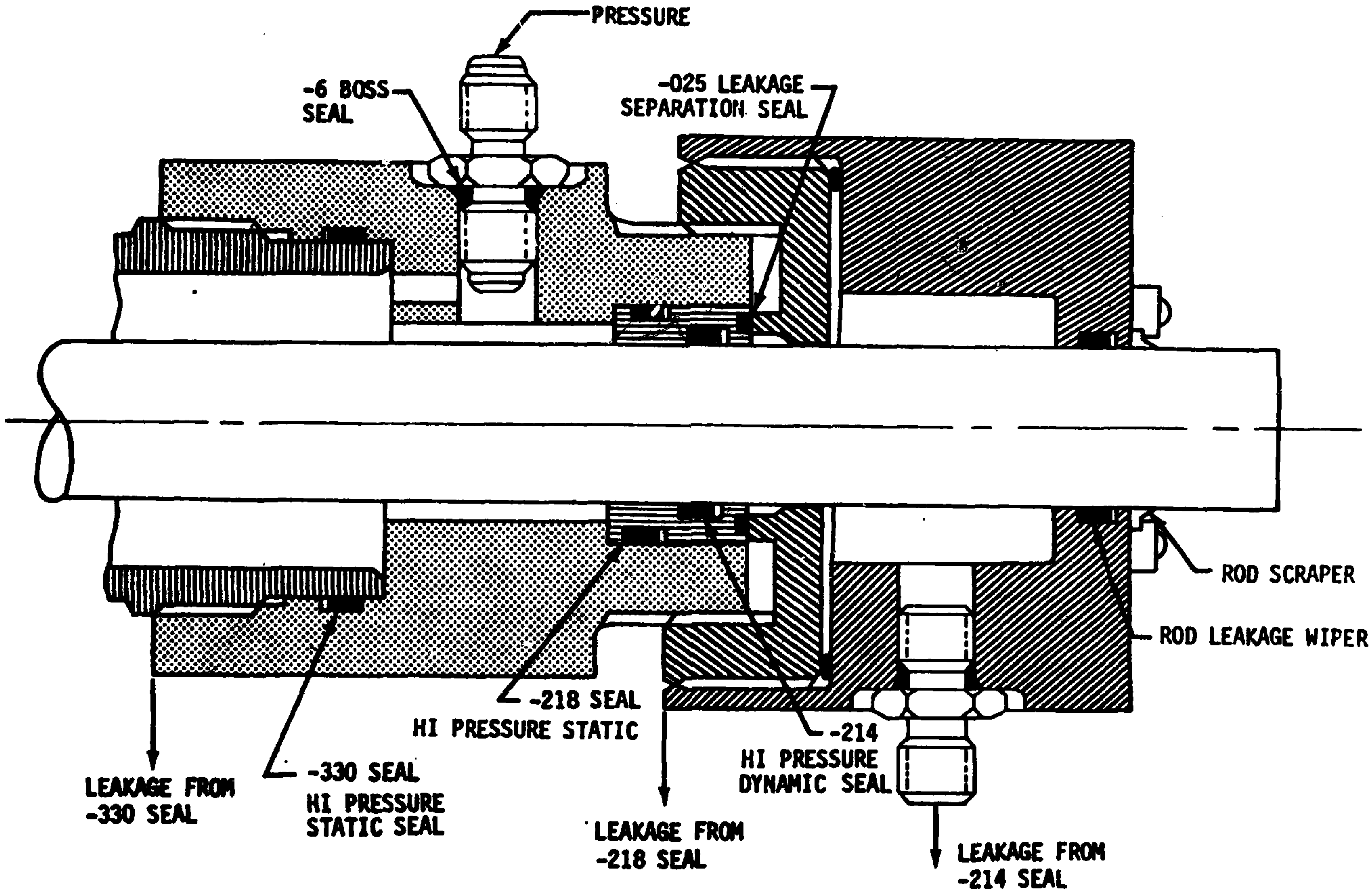


Figure 23. Military-standard seal test configuration

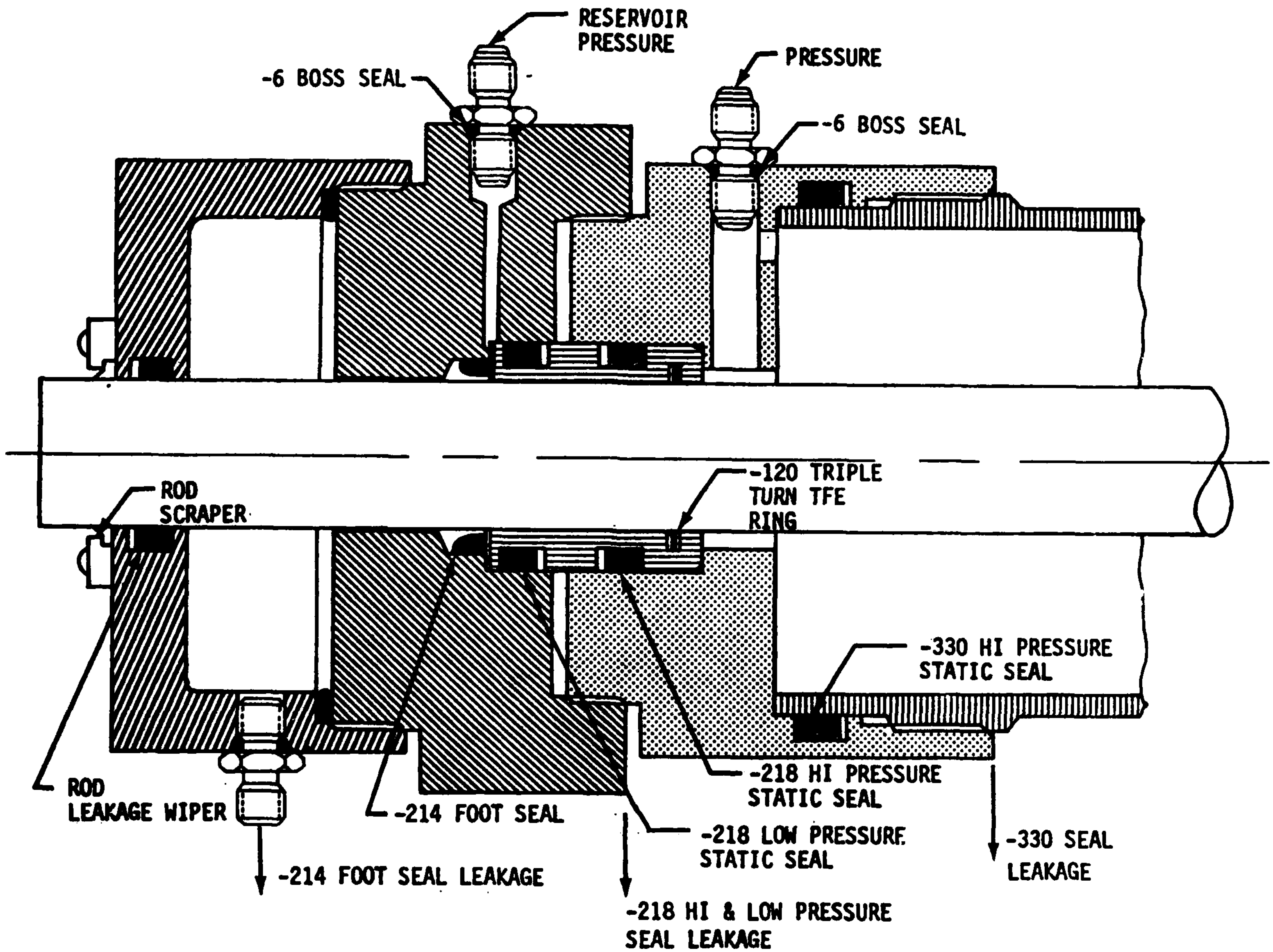


Figure 24. Boeing B-52 two-stage seal test configuration

.095+.005 inch width. The second-stage (low-pressure) seal was a standard Boeing BACS11AA-214 TFE Foot Seal with an MS28775-214 O-ring for the test with MIL-H-5606 fluid and with an AS568-214 size PNF O-ring for the AO-8 fluid test.

3.1.3.3 Diametral Static Seals

The test cylinder contained several military-standard static seals of both the internal and external configurations as shown in Figures 23 and 24. The seals were retained in single-backup-ring-width grooves per MIL-G-5514. Here again, MS28775 nitrile O-rings were used with MIL-H-5606 fluid and PNF O-rings with the AO-8 fluid. MS28782 dual-turn backup rings were used with both fluids, except for the cylinder end cap seals tested with the MIL-H-5606 fluid which used MS28774-330 single-turn backups.

3.1.4 Test Conditions

The seal testing was conducted per a technical plan developed to cover the testing requirements of MIL-P-25732B in the basic test phase, and those established by Boeing for the additional test phases. A summary of the test conditions is listed in Table 20 which compares the selected conditions and those specified in MIL-P-25732. Arrows point out testing similarity. The leakage acceptance criteria for MIL-P-25732 are also listed.

Figure 25 presents a temperature profile for the dynamic seal testing. The high-pressure cycling wave form is shown in the oscillograph trace of Figure 26. The pressure rise rate, calculated from the high-speed oscillograph trace of Figure 27 was found to be approximately 100,000 psi per second.

Dynamic seal testing generally requires seal leakage as the primary acceptance criteria. Secondary considerations are seal friction, and post-test seal and interface conditions. Leakage from several seal configurations and sizes, as shown in Figures 23 and 24, was collected in a graduated vial. The cummulative leakage was recorded at the end of each stage of testing. The seal friction was measured each time the test cylinder reached room temperature. The physical condition of the seals and their interfaces were visually examined and photographed at the end of testing.

In the additional test phase of the MIL-H-5606/nitrile testing, the 100,000 1/4-inch-stroke cycling was inadvertantly run with the pressure cycling from 3,000 psi to zero instead of continuously applied as specified in the plan. Although more severe, the pressure cycling did not result in any significant seal degradation.

3.1.5 Test Fluids

The standard petroleum-base hydraulic fluid, per MIL-H-5606C, was manufactured by Bray Oil Company as their Brayco Micronic 756E. The nonflammable fluid was manufactured by Halocarbon Products Corporation as their AO-8 fluid batch number 2877.

3.1.6 Test Results

The initial series of tests was run with the MIL-H-5606/nitrile

TABLE 20. TEST PROCEDURE SUMMARY

Boeing Tech. Plan

Friction @ RT & 0 psi
72 hrs @ 275F & 0 psi
1 hr @ 275F & 3K psi
10-4 in & 3K psi cycling @ 275F
10-4 in & 50 psi cycling @ 275F
1 hr @ 275F & 5 psi
Friction @ RT & 0 psi
Pressurize to 3K psi
24 hrs @ -65F & 10 psi
Translate rod*
10-4 in & 50 psi cycling @ -65F
Translate rod*
10-4 in & 3K psi cycling @ -65F
Translate rod*
1 hr @ -65F & 3K psi
Friction @ RT & 0 psi
10K-4 in & 3K psi cycling @ 275F
Friction @ RT & 0 psi
Pressurize to 3K psi
24 hrs @ -65F & 10 psi
Translate rod*
10-4 in & 50 psi cycling @ -65F
Translate rod*
10-4 in & 3K psi cycling @ -65F
Translate rod*
1 hr @ -65F & 3K psi
Friction @ RT & 0 psi
100K-¼ in cycling @ 3K psi & 225F
Friction @ RT & 0 psi
10K-4 in & 3K psi cycling @ 225F
Overnite @ 225F & 3K psi
10K-4 in & 3K psi cycling @ 225F
Overnite @ 225F & 3K psi
10K-4 in & 3K psi cycling @ 225F
Overnite @ 225F & 3K psi
10K-4 in & 3K psi cycling @ 225F
Overnite @ 225F & 3K psi
10K-4 in & 3K psi cycling @ 225F
Overnite @ 225F & 3K psi
10K-4 in & 3K psi cycling @ 225F
Overnite @ 225F & 3K psi
Friction @ RT & 0 psi
Pressurize to 3K psi
24 hr @ -65F & 10 psi
Translate rod*
10-4 in & 50 psi cycling @ -65F
Translate rod*
10-4 in & 3K psi cycling @ -65F
Translate rod*
1 hr @ -65F & 3K psi
Friction @ RT & 0 psi

MIL-P-25732 (Allowed Leakage)

72 hrs @ 275F
Pressurize to 1500 psi
24 hrs @ -65F & 10 psi
Translate rod*
10-4 in & 50 psi cycling @ -65F
Translate rod*
10-4 in & 1500 psi cycling @ -65F
Translate rod*
1 hr @ -65F & 1500 psi
— Allowed Leakage: 1 cc/ring
12K-4 in & 1500 psi cycling @ 275F
144 hrs @ 160F
1 hr: RT to 160F @ 1500 psi
24 hrs @ 160F & 1500 psi
10-4 in cycling @ 160F & 1500 psi
10-4 in cycling @ 160F & 5 psi
1 hr @ 160F & 10 psi
— Allowed Leakage: .15 cc/ring
3 hrs @ 160F & 1500 psi
20 hrs: 160F to RT @ 1500 psi
— Allowed Leakage: .10 cc/ring
Breakout friction
Pressurize to 1500 psi
72 hrs @ -65F & 0 psi
Translate rod*
10-4 in & 50 psi cycling @ -65F
Translate rod*
1 hr @ -65F & 1500 psi
— Allowed Leakage: 1 cc/ring
10K-4 in cycling @ 120F & 1500 psi
Overnite @ 10 psi & 120F
10K-4 in cycling @ 120F & 1500 psi
Overnite @ 10 psi & 120F
10K-4 in cycling @ 120F & 1500 psi
Overnite @ 10 psi & 120F
10K-4 in cycling @ 120F & 1500 psi
Overnite @ 10 psi & 120F
10K-4 in cycling @ 120F & 1500 psi
Overnite @ 10 psi & 120F
10K-4 in cycling @ 120F & 1500 psi
Overnite @ 10 psi & 120F
10K-4 in cycling @ 120F & 1500 psi
— Allowed Leakage: 35 cc/ring

1 Basic Test Phase
2 Additional Test Phase

* up & down, and side-to-side

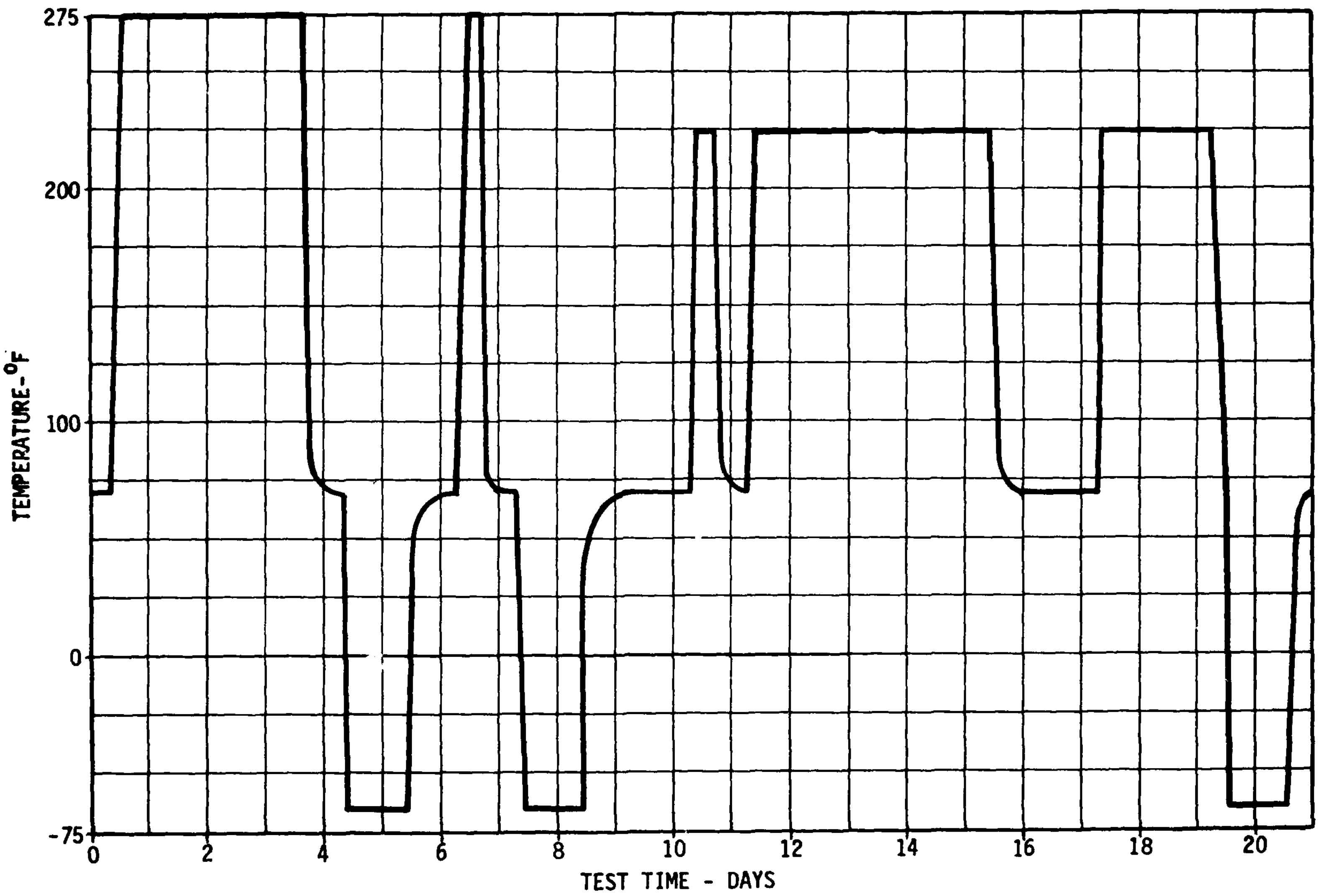


Figure 2[illegible]. Test seal temperature profile

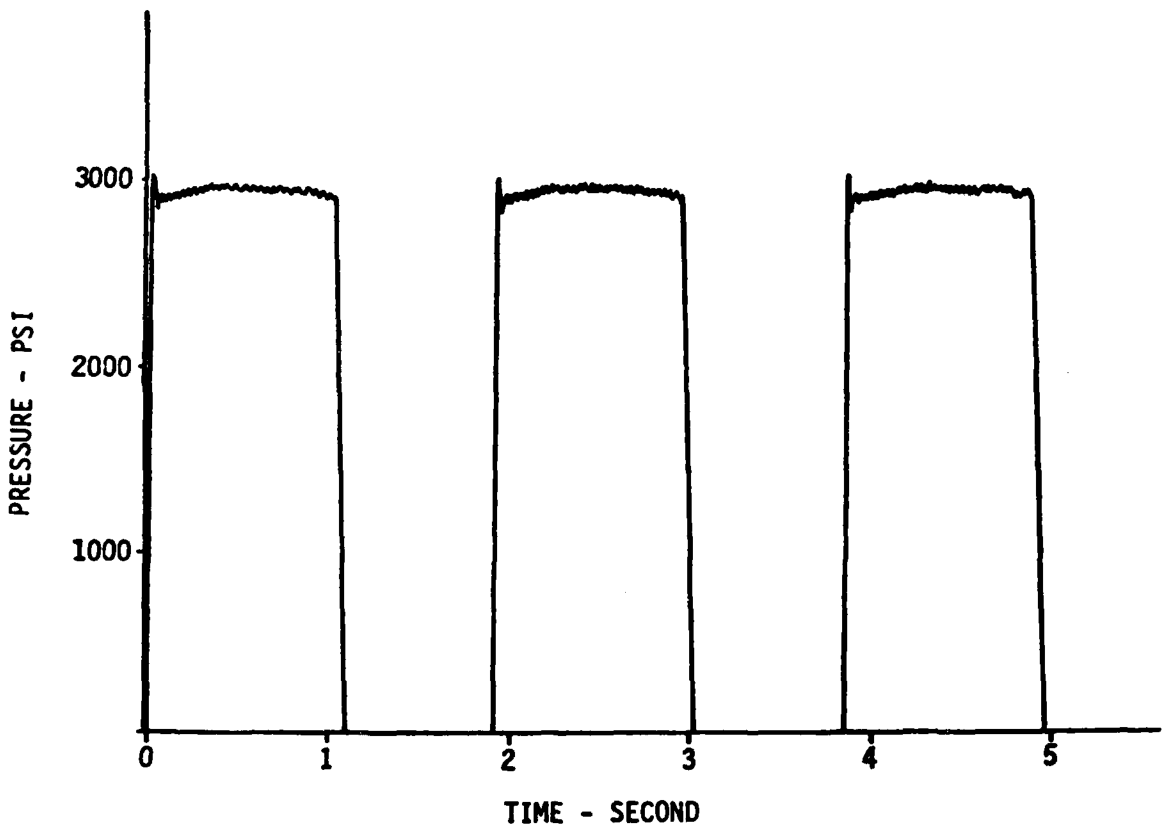


Figure 26. High-pressure cycling oscillograph trace

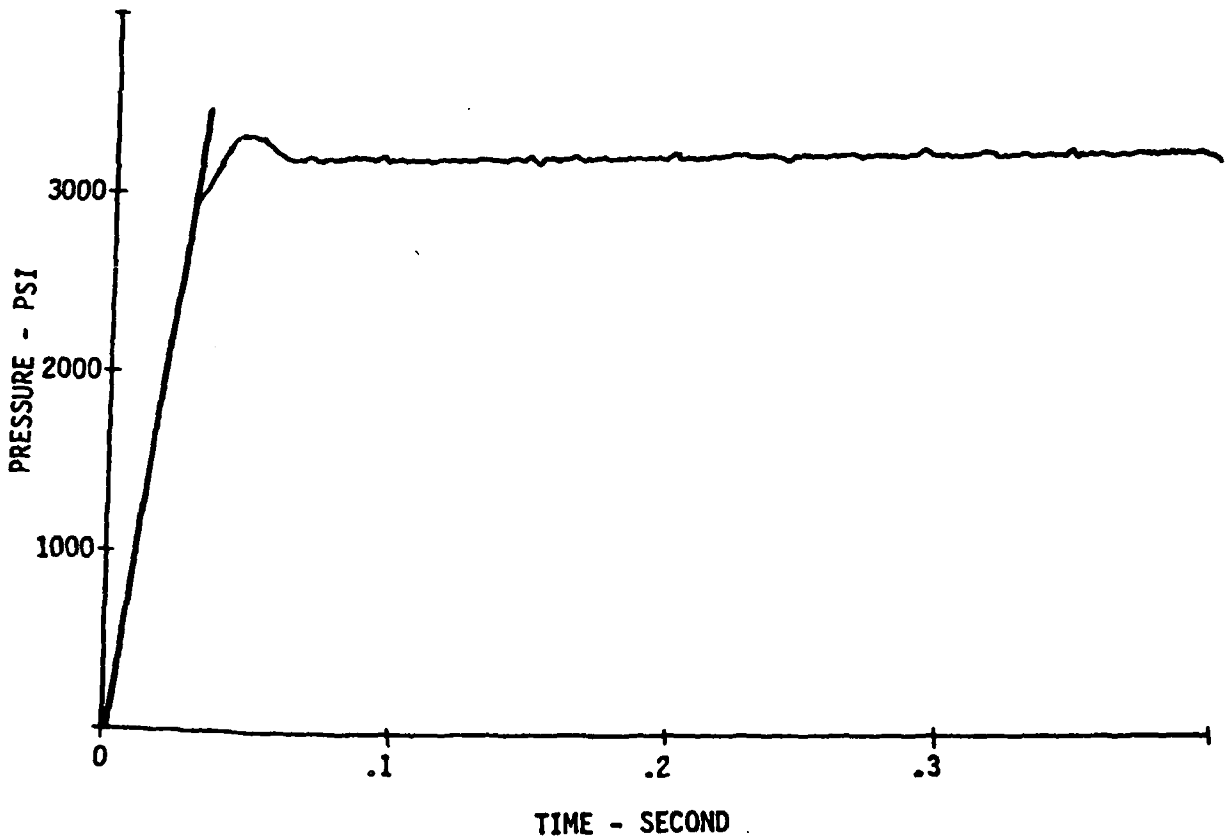


Figure 27. Pressure rise rate oscillograph trace

fluid/seal combination, and was considered a reference test so that acceptance criteria could be developed for the AO-8/PNF fluid/seal test. The accumulated leakage and the condition of the seals and mating parts after the tests were as follows.

3.1.6.1 Leakage - MIL-H-5606/Nitrile Seals and AO-8/PNF Seals

As shown in Table 21, seal leakage during the test did not exceed the requirements of MIL-P-25732B. However, at the end of the additional testing phase of the MIL-H-5606/nitrile reference test, when the reservoir was depressurized to atmospheric pressure and the test cylinder allowed to warm up overnight, it was found that the two static MS28775-330 (2.5-inch OD) size O-rings had leaked an additional 11.4 and 20 milliliters respectively. In addition, it was found that the MS28775-218 (1.5-inch OD) size high-pressure static seal and the MS28775-214 (1.0-inch ID) size high-pressure dynamic seal in the standard seal gland had leaked an additional 3.9 and 8.9 milliliters respectively. The absence of fluid pressure during the overnight warmup period apparently de-energized the nitrile seals, allowing the excessive leakage.

TABLE 21. SEAL LEAKAGE SUMMARY

	Boeing B-52 Seal Gland Leakage (Milliliters)			Military Standard Gland Leakage (Milliliters)		
Fluid/Seal Combination	-214 Foot Seal	-218 Hi&Low-Pr Static	-330 Hi-Pres Static	-330 Hi-Pres Static	-218 Hi-Pres Static	-214 Hi-Pres Dynamic
MIL-H-5606/ Nitrile	0	0.1	0.9	0.8	0	2.1
AO-8/PNF	Trace	Trace	1.2	0	0	0.1 plus*

*An estimated 1-2 ml. of fluid passed thru the rod wiper and puddled on the insulated box bottom.

3.1.6.2 Teardown Inspection - MIL-H-5606/Nitrile Seal Test

Inspection following the MIL-H-5606/nitrile seal reference test indicated little evidence of legitimate seal and/or interface deterioration. Most of the seals were somewhat harder than new O-rings and contained evidence of normal deformation. The appearance of the metallic parts was unchanged by the test.

3.1.6.2.1 Rod

The test cylinder rod showed light longitudinal scratches and burnishing in the dynamic seal interface areas. None of these visual marks in the chrome plating could be detected by feel or fingernail.

3.1.6.2.2 Military-Standard Rod Seal

The standard configuration MS28775-214 dynamic rod seal (Figure 28) had taken some permanent set, and light scratches were found on the inside diameter. These scratches were caused by the stroking rod; and, since no large amount of black silt was encountered, were considered as normal wear. The nitrile O-ring permanent set was marginally excessive since the seal allowed 8.9 milliliters of leakage as the test was shutdown at -65F. The MS28782-19 TFE two-turn backup ring in this gland had a moderate extrusion lip on approximately 220 degrees of the inside diameter circumference.

3.1.6.2.3 B-52 First-Stage Rod Seal

The Shamban P/N S12003-120-3 triple-turn high-pressure dynamic rod seal (Figure 29) utilized in the B-52 servoactuator seal system configuration end of the test cylinder (shown in Figure 24), leaked rather extensively throughout the test. A great deal of difficulty at installing the seal in its gland was encountered during the test cylinder buildup. Therefore, it was felt that the TFE ring was probably damaged during installation. The leakage was estimated by a computation using the power transfer cylinder's time-to-bottom after refilling. The highest leakage rates occurred during the high-pressure low-temperature soak, and reduced to near zero at higher temperatures. The leakage was internal to the test system; that is the fluid returned to the reservoir.

First Cold Soak	(@ 3000 psi)	2.9 ml/min.
Second Cold Soak	(@ 3000 psi)	7.5 ml/min.
Third Cold Soak	(@ 3000 psi)	2.5 ml/min.

The surfaces had extensive black residue packed into the TFE material but little evidence of extrusion. Some evidence of inside diameter wear was discernable.

3.1.6.2.4 B-52 Second-Stage Rod Seal

The Boeing P/N BACS11AA-214 TFE Foot Seal (Figure 30) showed very little testing deterioration. A slight extrusion of the low-pressure inside-diameter corner and black particles in the inside-diameter grooves were the only discernable changes.

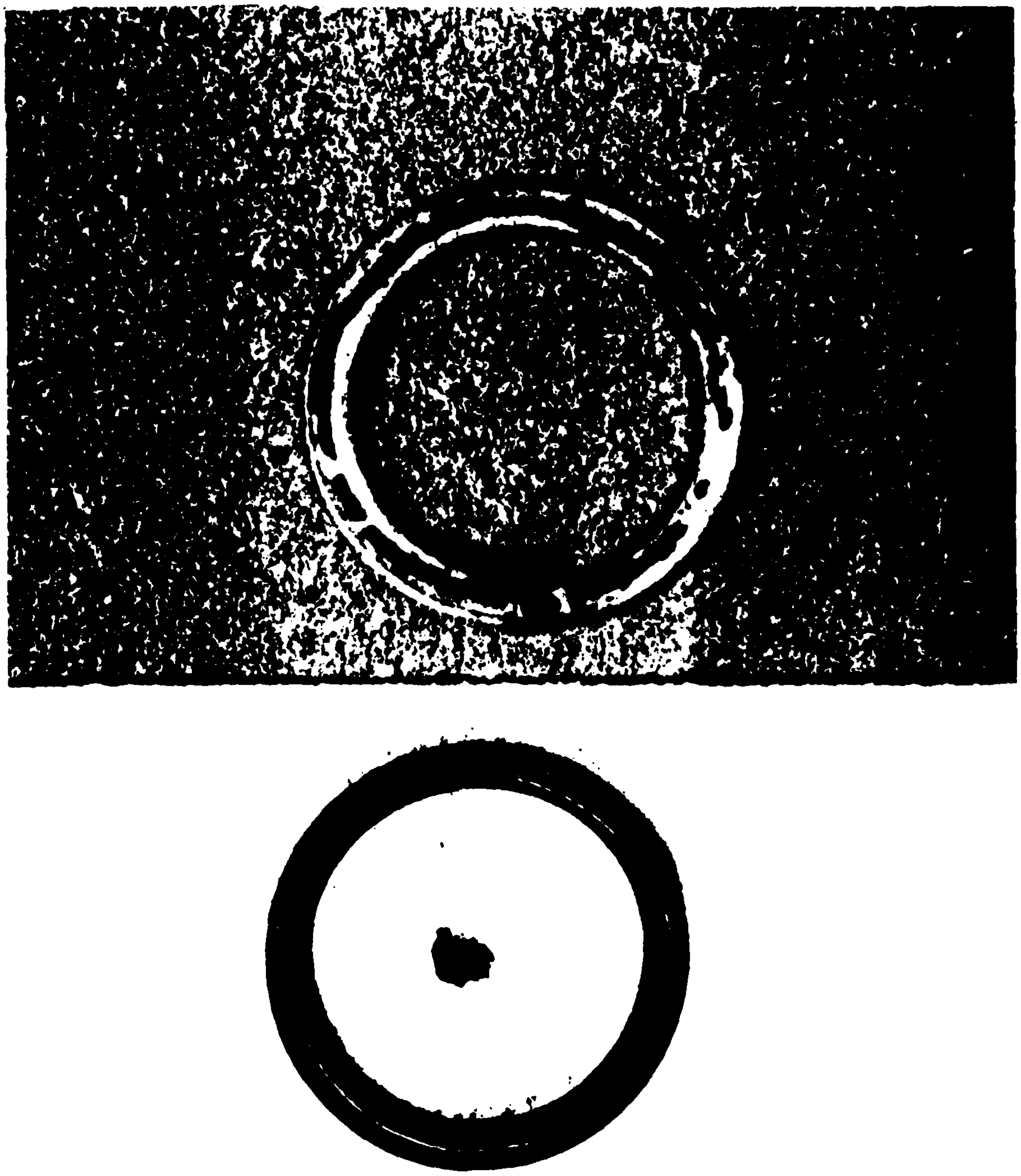


Figure 28. Military-standard rod seal (MIL-H-5606 fluid)

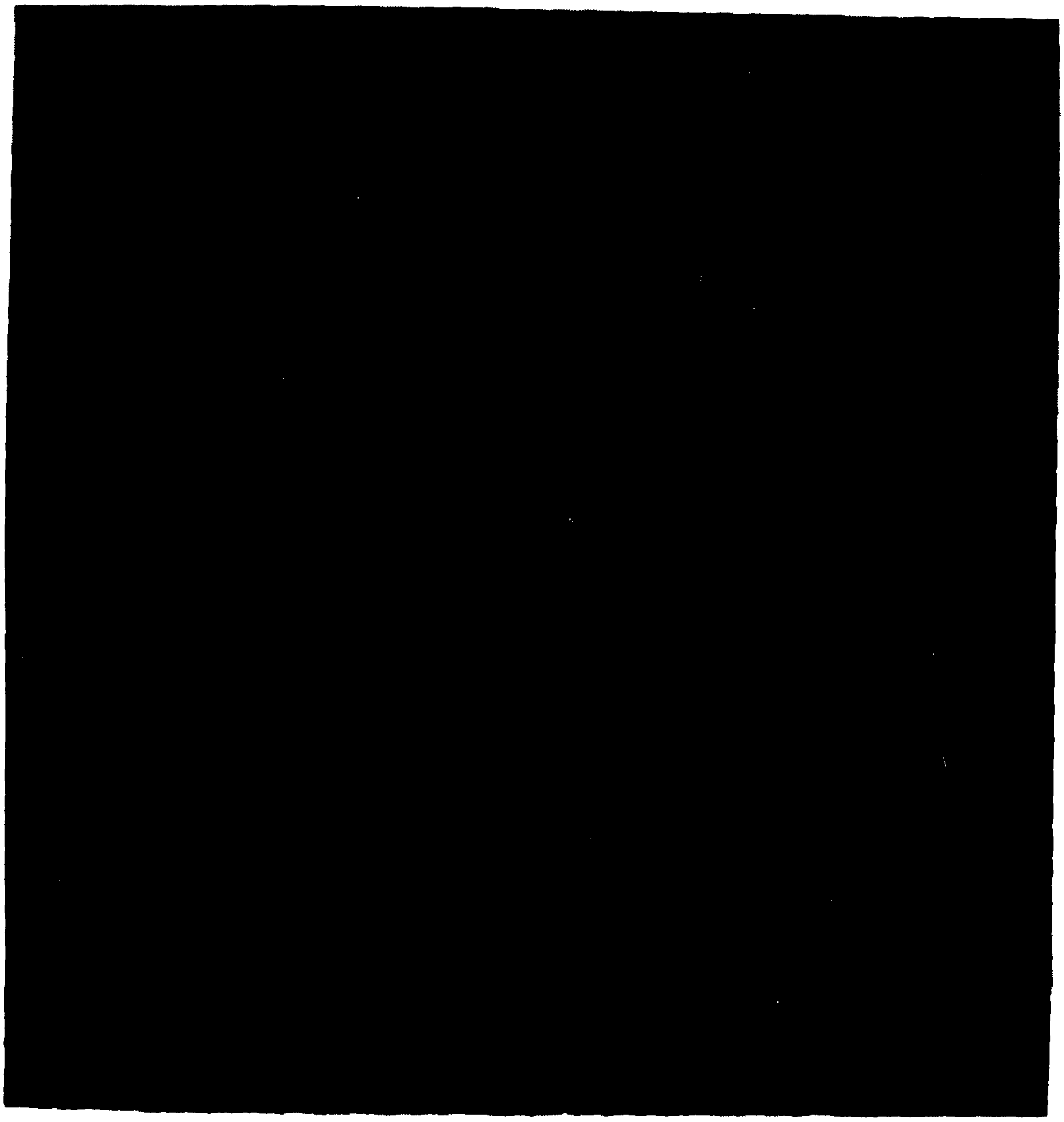


Figure 29. First-stage rod seal (MIL-H-5606 fluid)

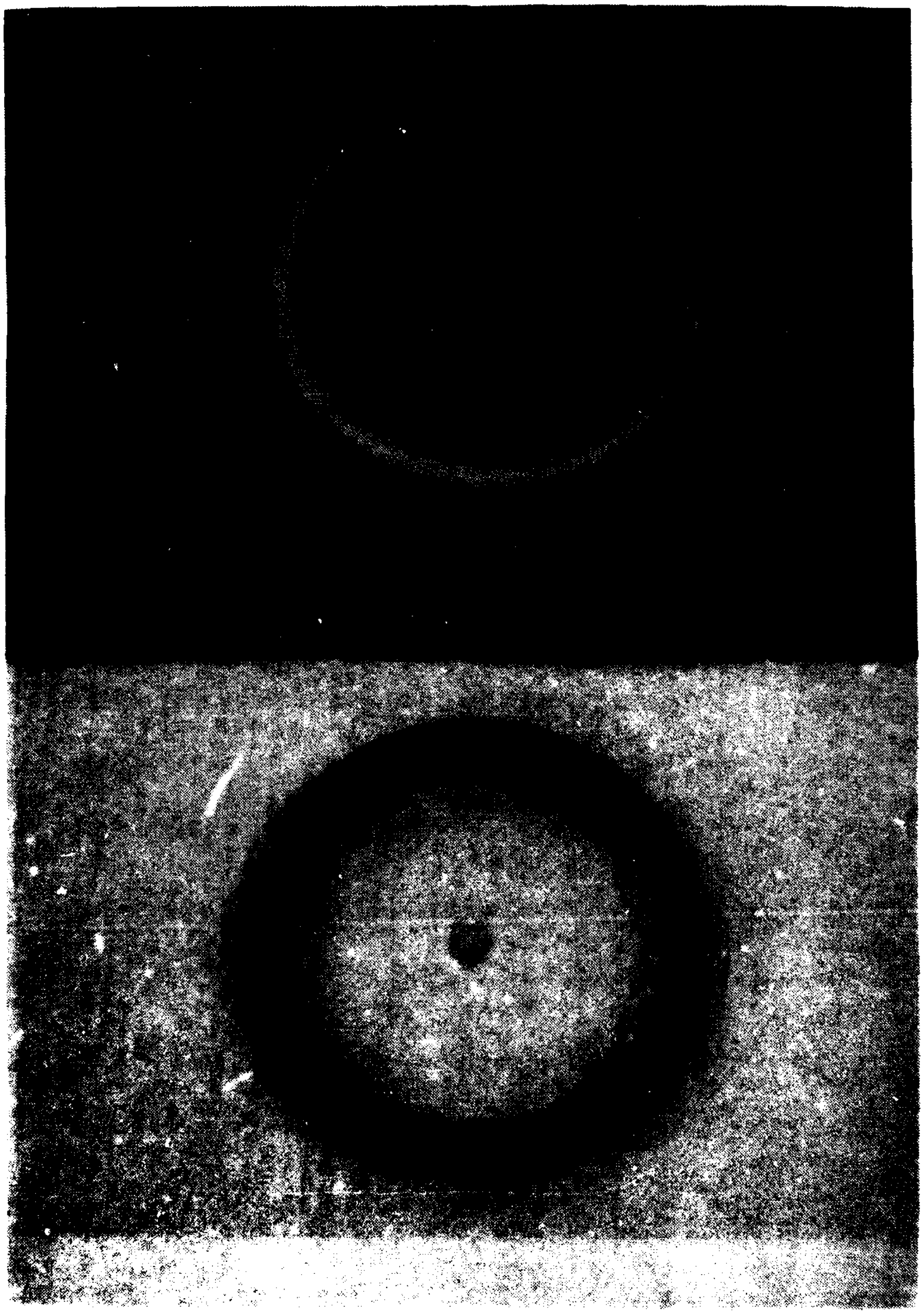


Figure 30. Foot seal (MIL-H-5606 fluid)

The MS28775-214 O-ring used with the Foot Seal had the normal permanent deformation associated with this installation. Also, the Foot Seal O-ring extruded into the four fluid slots in the rod bearing. This probably occurred because of elastomer swelling. A design fix utilizing a thin metal washer was used in the AO-8 test to prevent a similar occurrence.

3.1.6.2.5 Military-Standary External High-Pressure Static Seals

At the standard seal configuration end (Figure 23) of the test cylinder, the high pressure static seal gland had a thick black residue in and around the groove. The MS28775-218 seal (Figure 31) was covered with the silt-like residue but when wiped clean showed no signs of wear. The seal's OD and ID molding flash ridges were visible. However, definite compression set had taken place in the O-ring, and the MS28782-23 dual-turn TFE backup ring had a small extrusion ridge for nearly the entire circumference. In summary, however, it could not be determined where the black particle residue originated.

The high-pressure static O-ring seal in the B-52 seal bearing had extensive extrusion and nibbling due to the inadvertant installation of the backup ring on the wrong (high-pressure) side of the O-ring. However, this extensive degradation (see Figure 32) allowed but 0.1 ml cummulative leakage during the complete test.

3.1.6.2.6 Military-Standard External Low-Pressure Static Seal

The MS28775-218 seal used at low (reservoir) pressure on the Foot-Seal bearing was installed in an external groove with no backup. The seal at the end of testing showed no deterioration.

3.1.6.2.7 Military-Standard Internal High-Pressure Static Seal

The two MS28775-330 sized high-pressure static O-rings (Figure 33) had some permanent set. Both O-rings developed a protruding bump corresponding to the bias cut in the MS28774-330 single-turn TFE backup rings.

3.1.6.2.8 Boss Seals

The two MS28778-6 high-pressure boss fitting seals (Figure 34) had the usual high deformation found in this type of installation. The frayed outside edge is indicative of excess seal volume, also normal in these seal installations.

3.1.6.3 Teardown Inspection - AO-8/PNF Seal Test

Inspection following the Halocarbon AO-8/Firestone PNF seal testing revealed a significant discoloration on most of the 4340 steel components and indications of seal adhesion in the seal grooves. The test cylinder had significant discoloration on most of the 4340 steel components (see Figures 35 and 36). The rusty coloration did not generally cover all surfaces, but most parts had some portion of their surface discolored. Also, the rusty coloration had variable shades; and, one part (Figure 37) had a distinct demarcation on one surface with heavy discoloration in one sector, but was unaffected in the remainder.

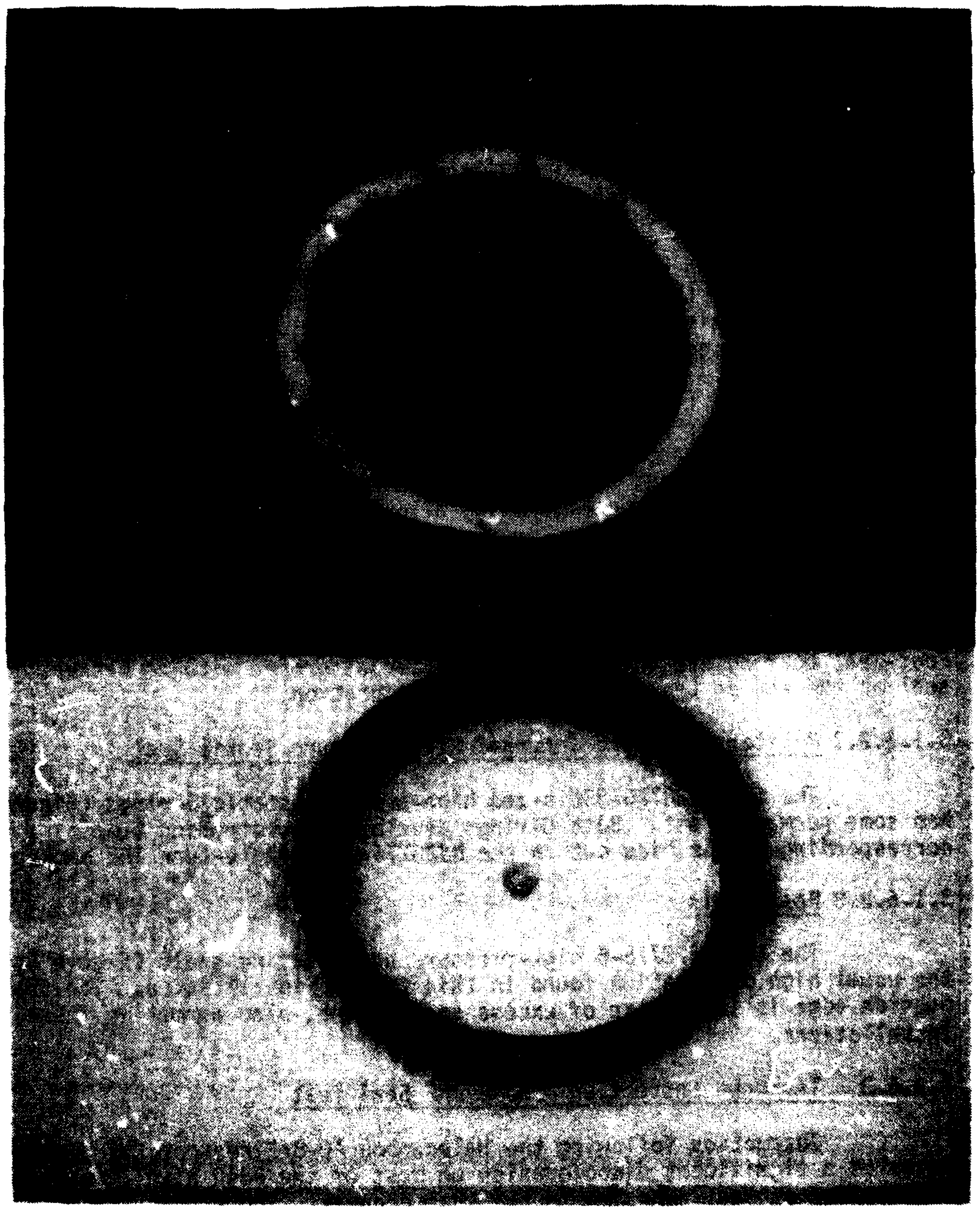


Figure 31. Military-standard high-pressure static seal (MIL-H-5606 fluid)

Figure 32. High-pressure static seal (two-stage seal bearing) (MIL-H-5606 fluid)

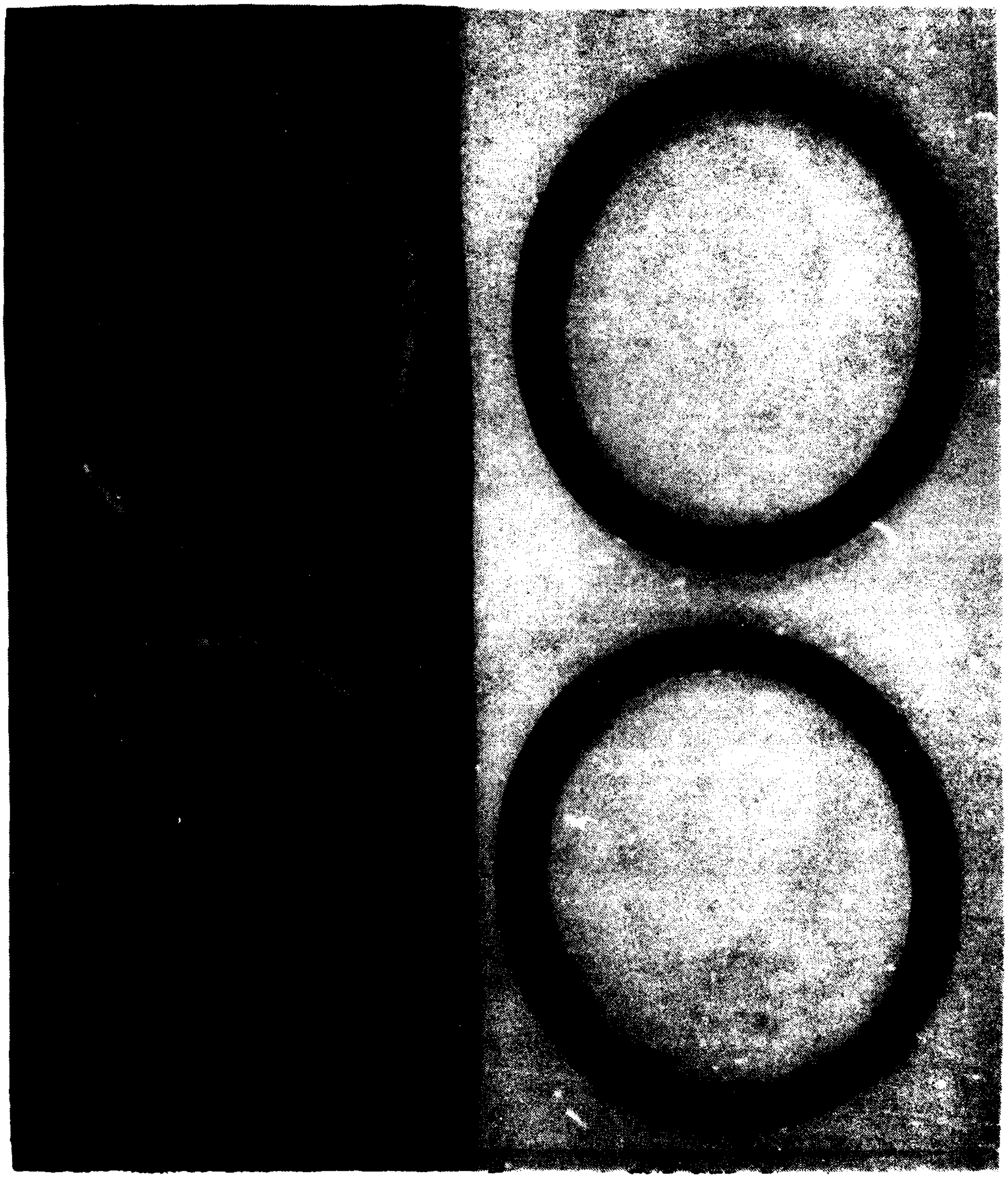


Figure 33. Military-standard internal static seals (MIL-H-5606 fluid)

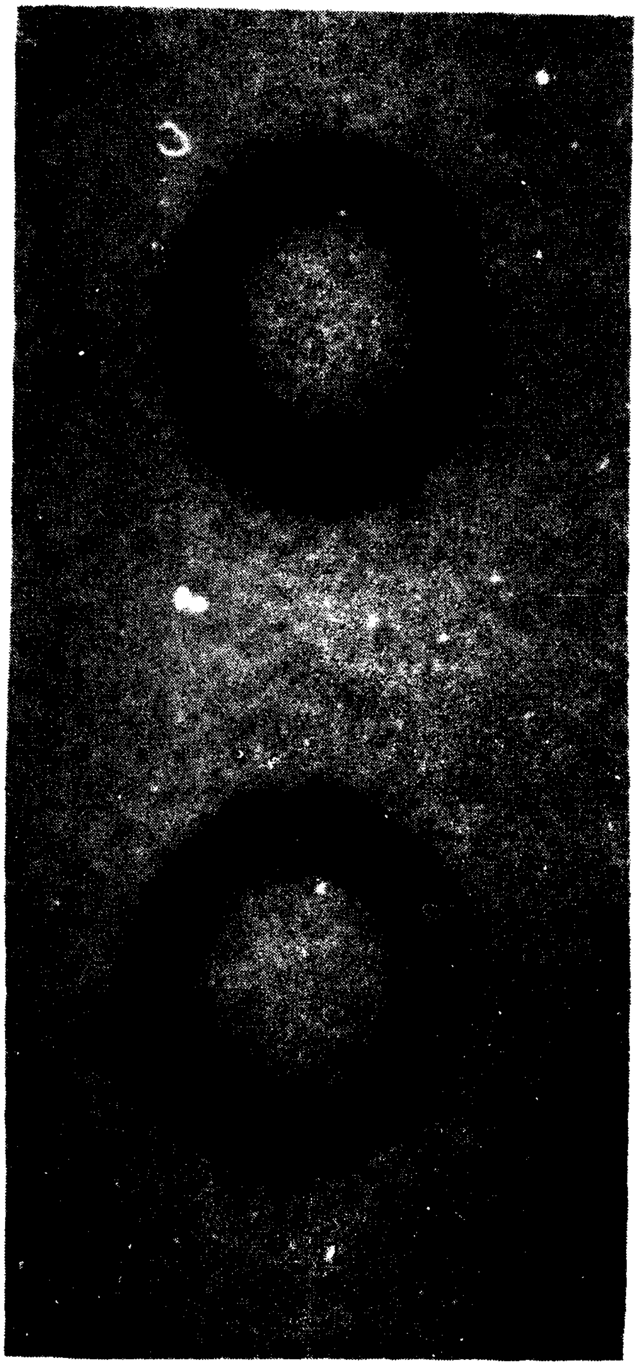


Figure 34. High-pressure boss fitting seals (MIL-H-5606 fluid)

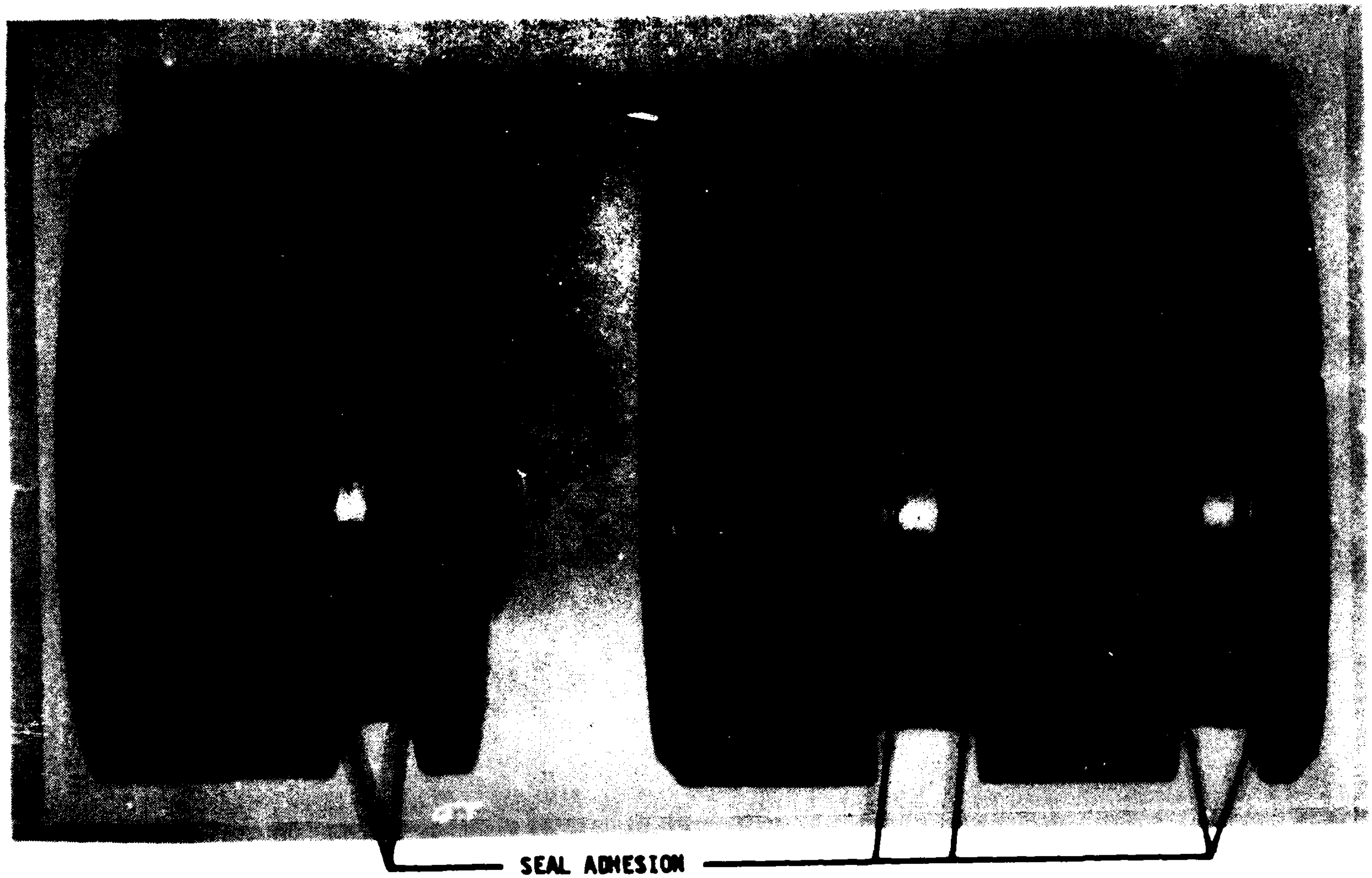


Figure 35. Rod bearings after AO-8 fluid test

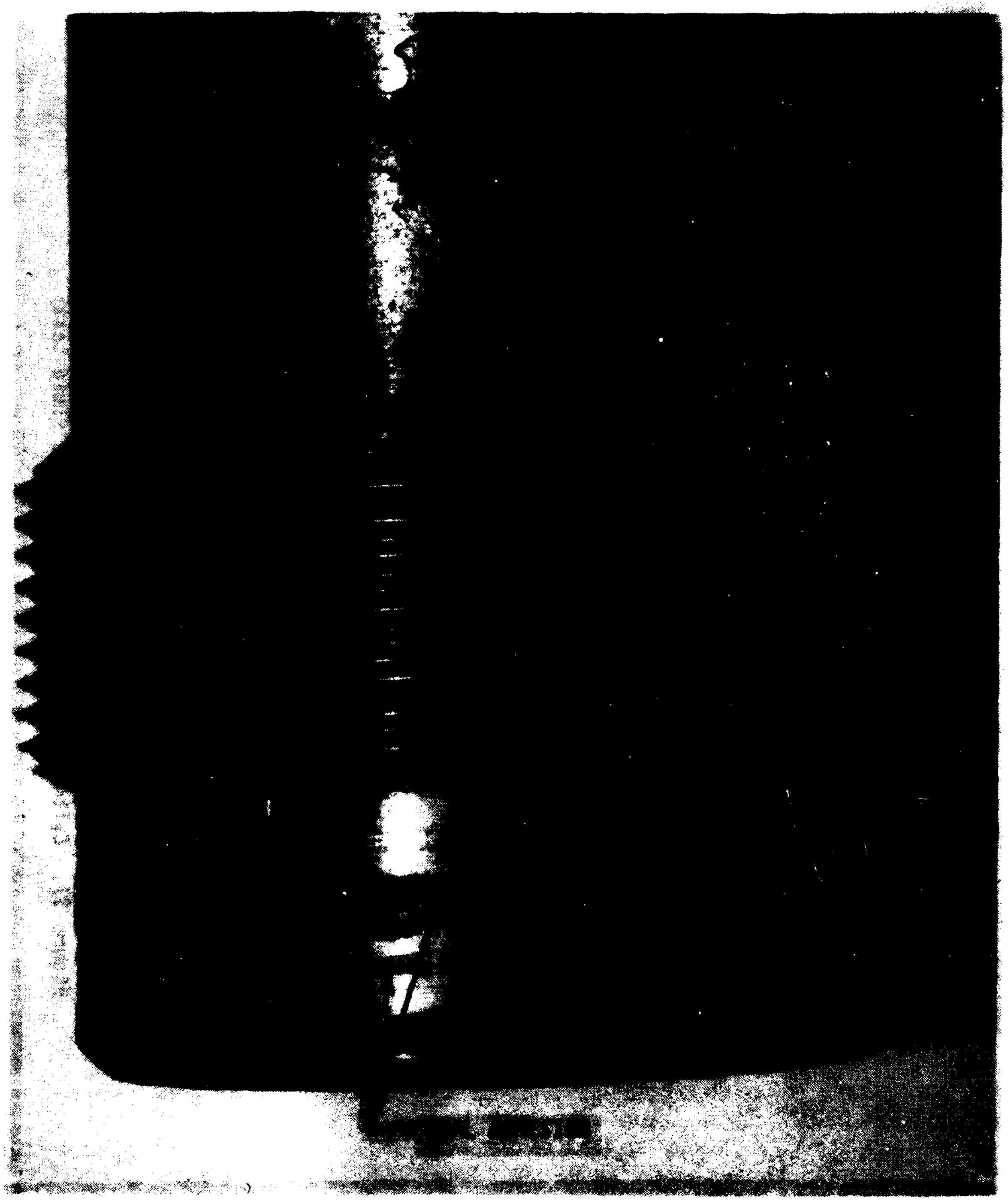


Figure 36. Cylinder at seal interface after AO-8 fluid test

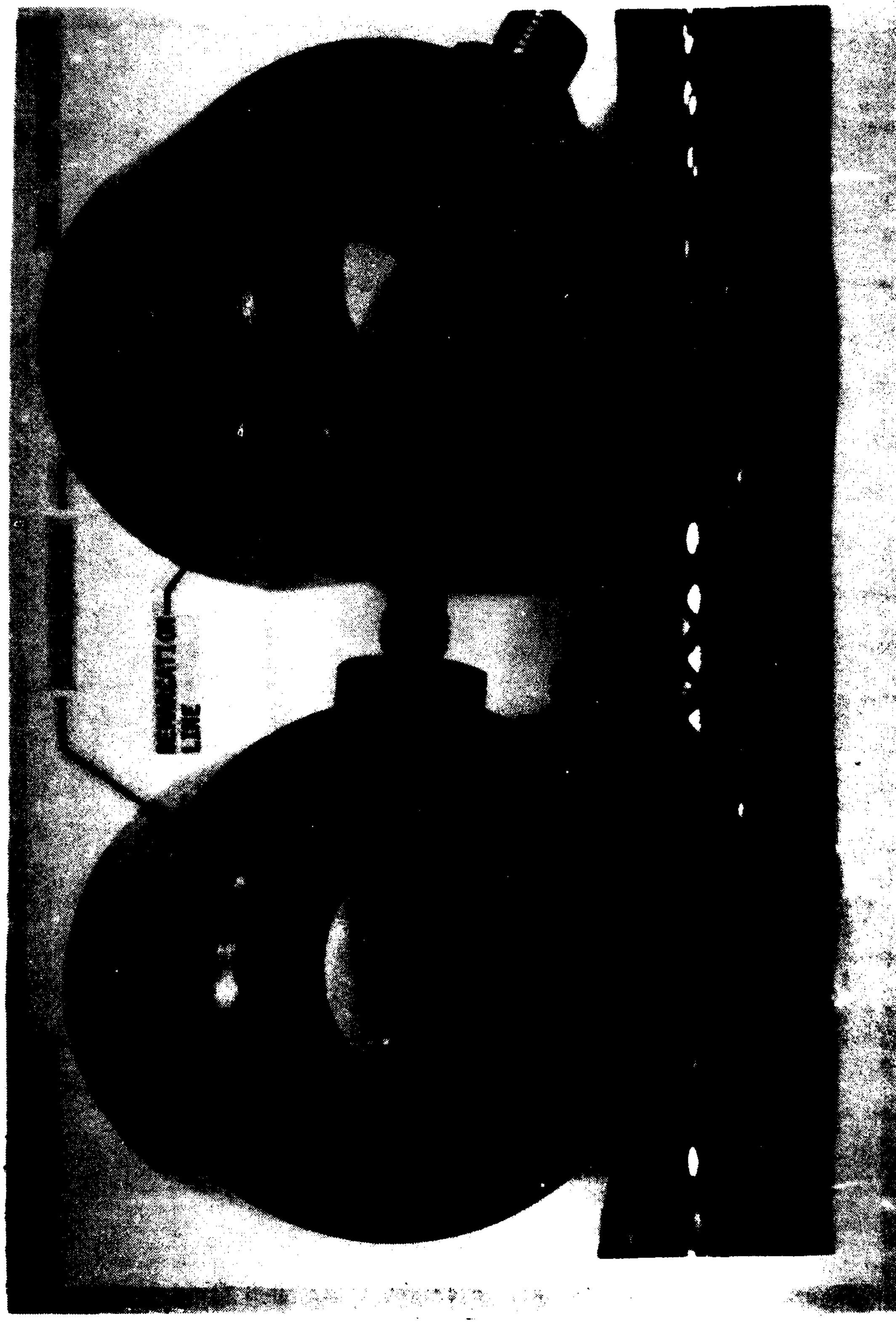


Figure 37. Cylinder and fittings after AO-8 fluid test

All of the PNF elastomer O-rings removed from the test cylinder were very soft and compliant with less than normal permanent deformation. Elastomer particles and chunks remained in most glands when the O-rings were removed. The fluid contained dark particles in suspension and had a light yellow-brown coloration.

3.1.6.3.1 Rod

The test cylinder's chrome-plated rod (Figure 38) had the normal polishing wear patterns, but no significant deterioration was visible.

3.1.6.3.2 Military-Standard Rod Seal

The standard-configuration AS568-214 size PNF dynamic rod seal (Figure 39) had a heavy coating of silt and had rather extensive extrusion-type damage on the outside diameter. The damage corresponded to a section in the backup ring that had significant dimensional change. The backup's TFE material had extruded, drawing out sufficient material to reduce its thickness and radial width. The inside diameter of the O-ring had worn due to the stroking action of the rod.

3.1.6.3.3 B-52 First-Stage Rod Seal

The TFE triple-turn dynamic rod seal (Shamban P/N S12003-120-3) shown in Figure 40, had a thin extrusion ridge on approximately 250° of the inside circumference.

3.1.6.3.4 B-52 Second-Stage Rod Seal

The TFE Foot Seal shown in Figure 41, (Boeing P/N BACS11AA-214) had a thin extrusion ridge on all of the inside circumference on both the atmosphere and fluid sides. The inside diameter surface had the normal light-wear pattern and the three grooves remained visible although filled with a black residue. The AS568-214 size PNF O-ring showed signs of less than normal deformation for this type of installation.

3.1.6.3.5 Military-Standard External High-Pressure Static Seals

The AS568-218 size high-pressure static seal (Figure 42) from the military-standard seal bearing, had some silt adhering to the fluid side of the seal and a significant number of larger elastomer particles in the groove.

The MS28782-19 TFE backup ring was a classic example of cold-flow extrusion. The ring had moderate to extensive extrusion at the O.D. and dimensional changes to its thickness and radial width.

The AS568-218 size high-pressure static seal (Figure 43) from the B-52 actuator configuration end had slight evidence of nibbling on the inside diamaeter. This damage would have occurred where the O-ring extruded under the backup ring. The backup ring showed evidence of moderate extrusion on the O.D. and of having a sight overlap of the bias cuts.

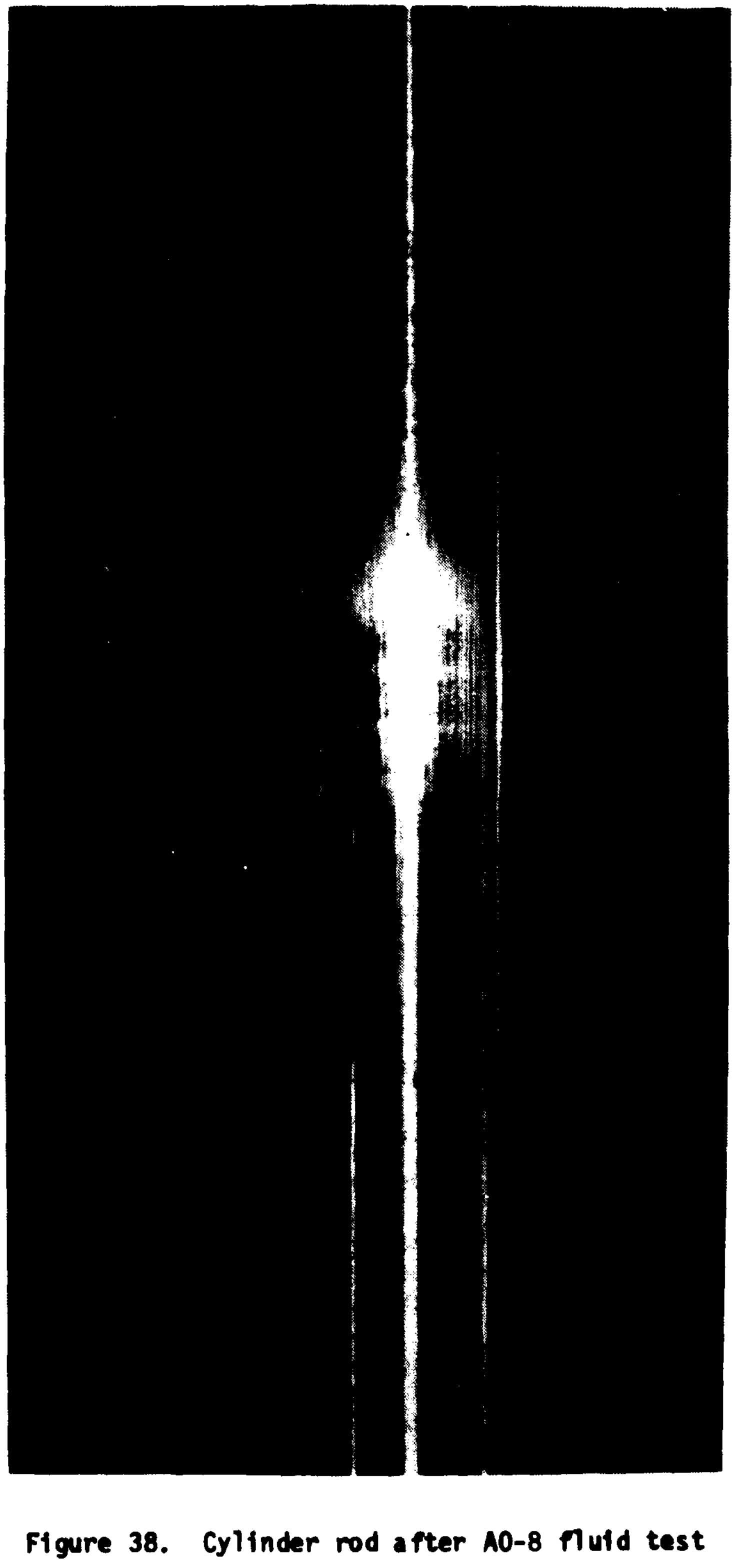


Figure 38. Cylinder rod after A0-8 fluid test

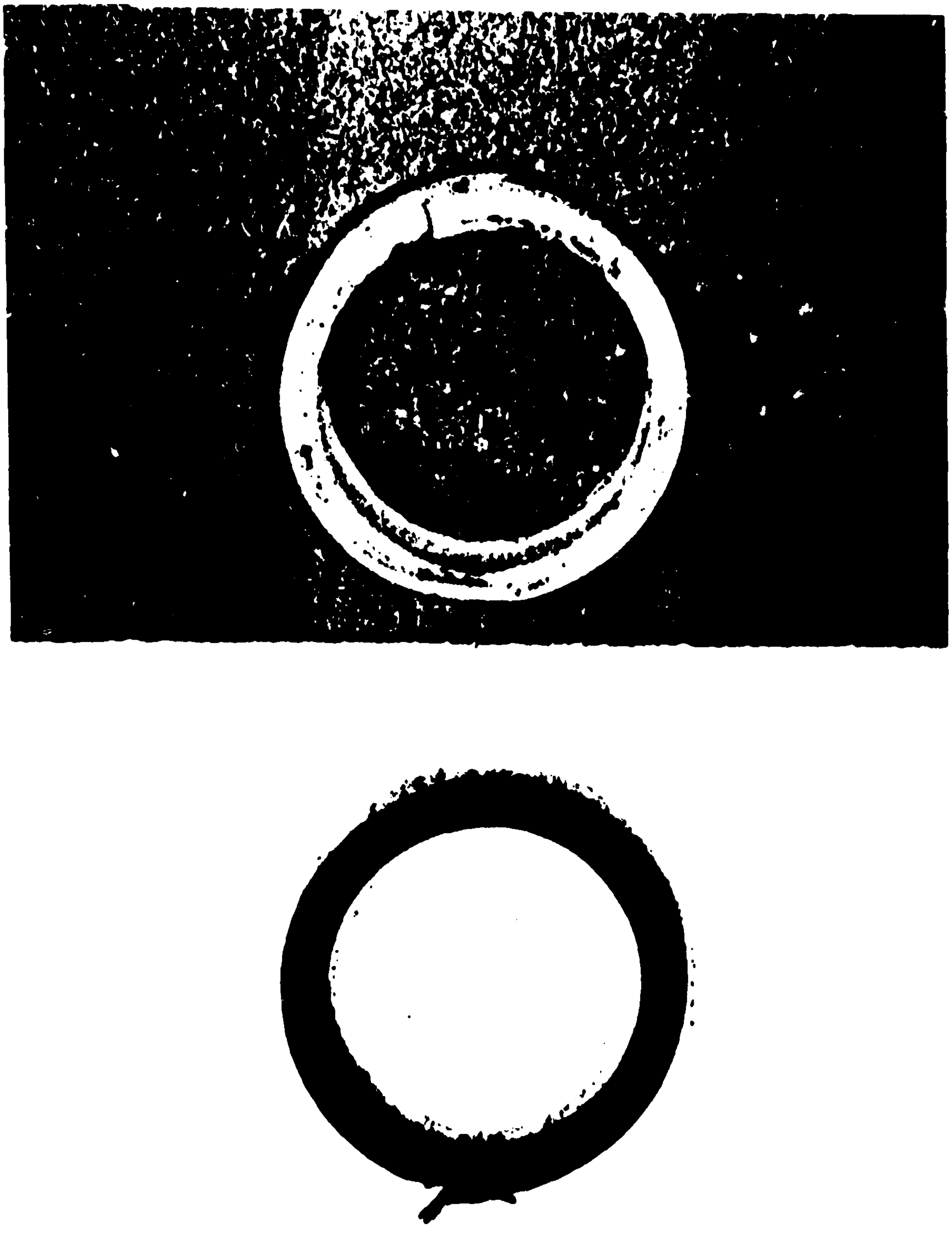


Figure 39. Military-standard dynamic seal (AO-8 fluid)

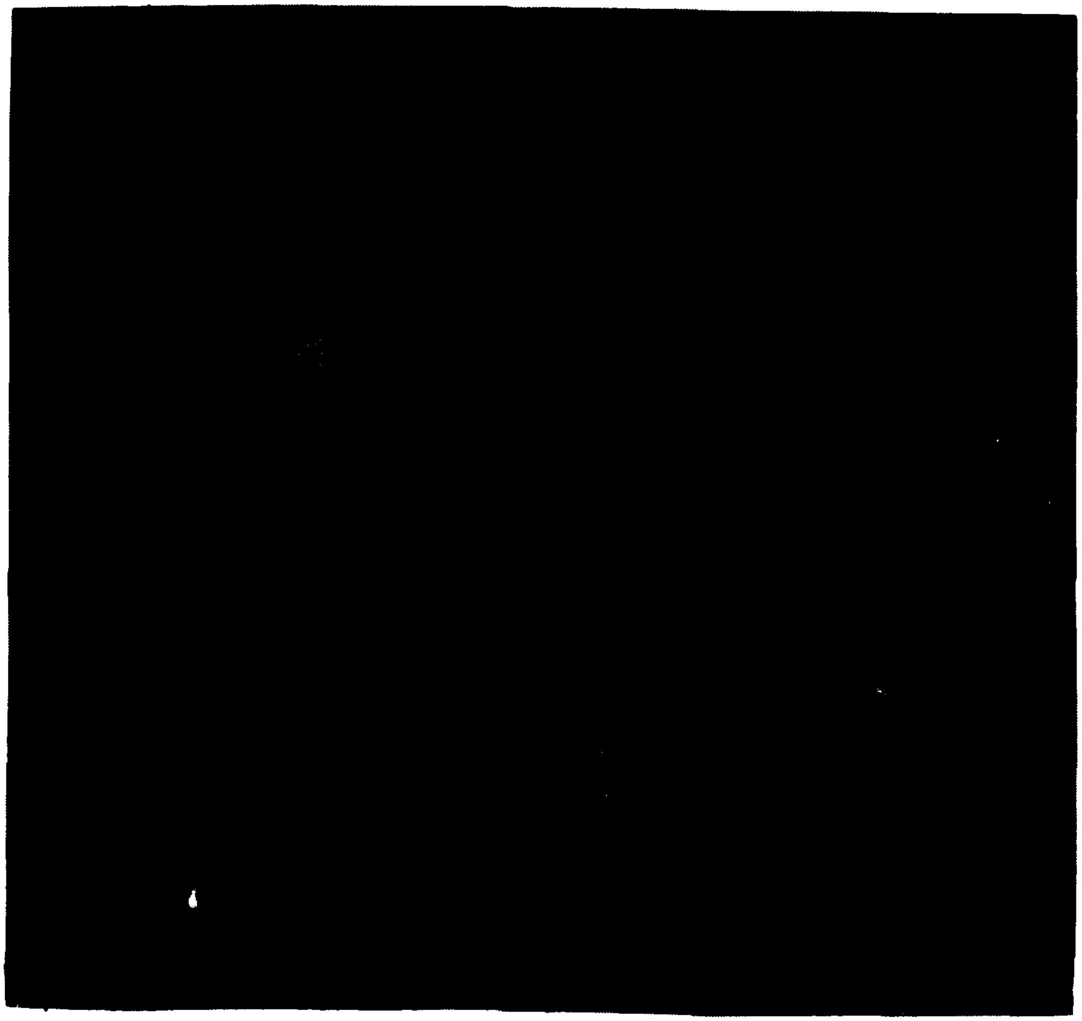


Figure 40. First-stage TFE rod seal (AO-8 fluid)

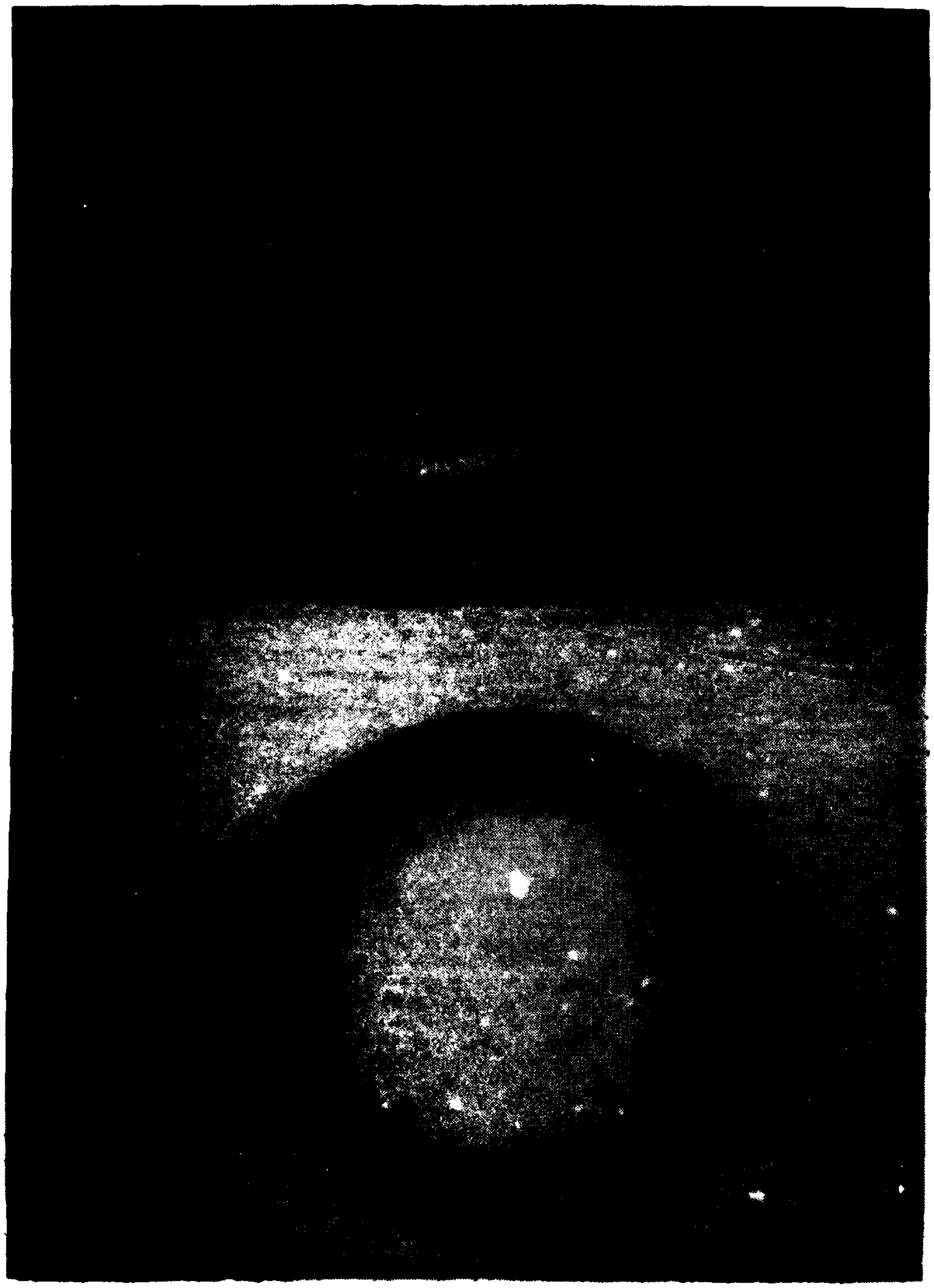


Figure 41. Test seal (AO-8 fluid)

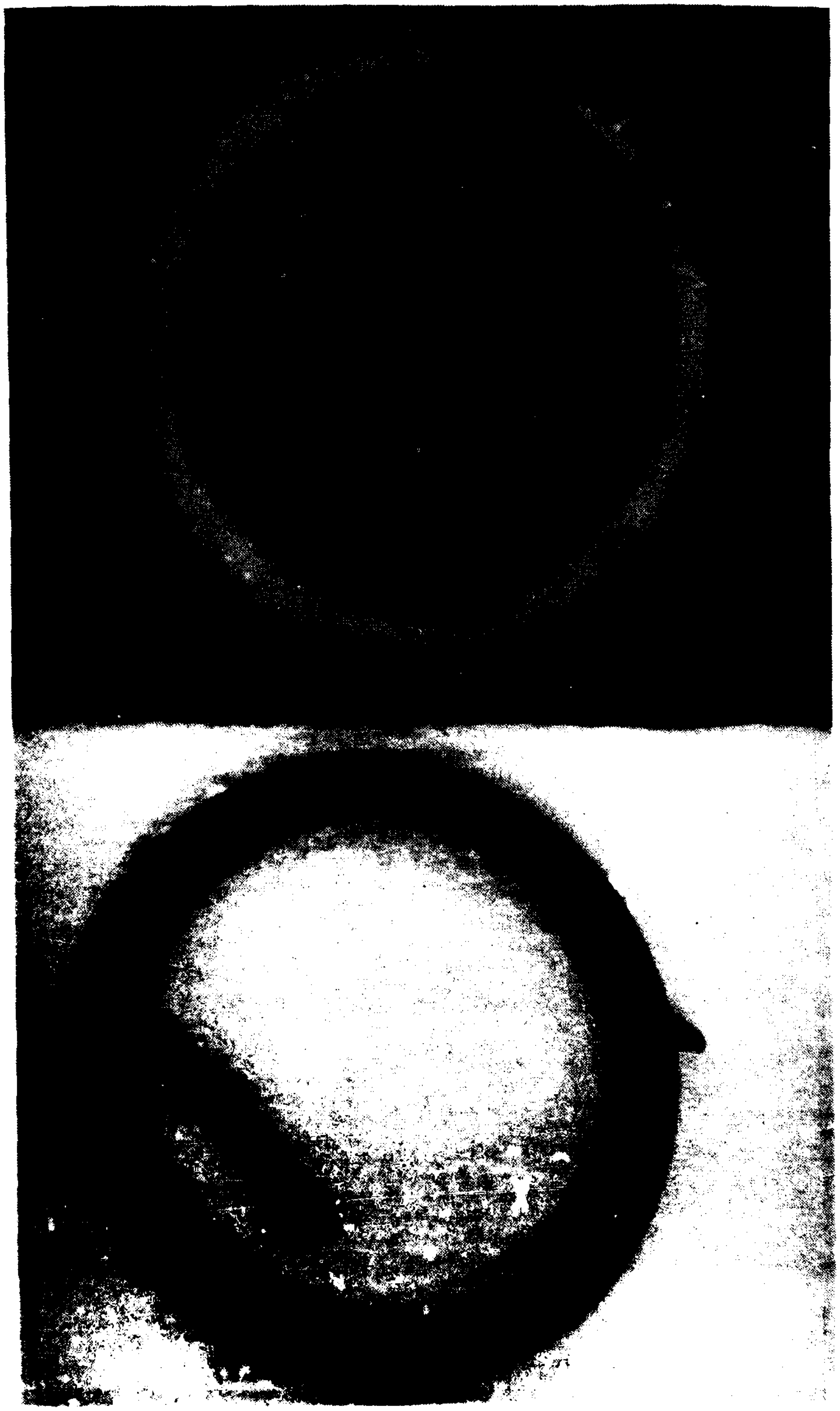


Figure 42. Military-standard high-pressure static seal, B-52 bearing (AO-8 fluid)

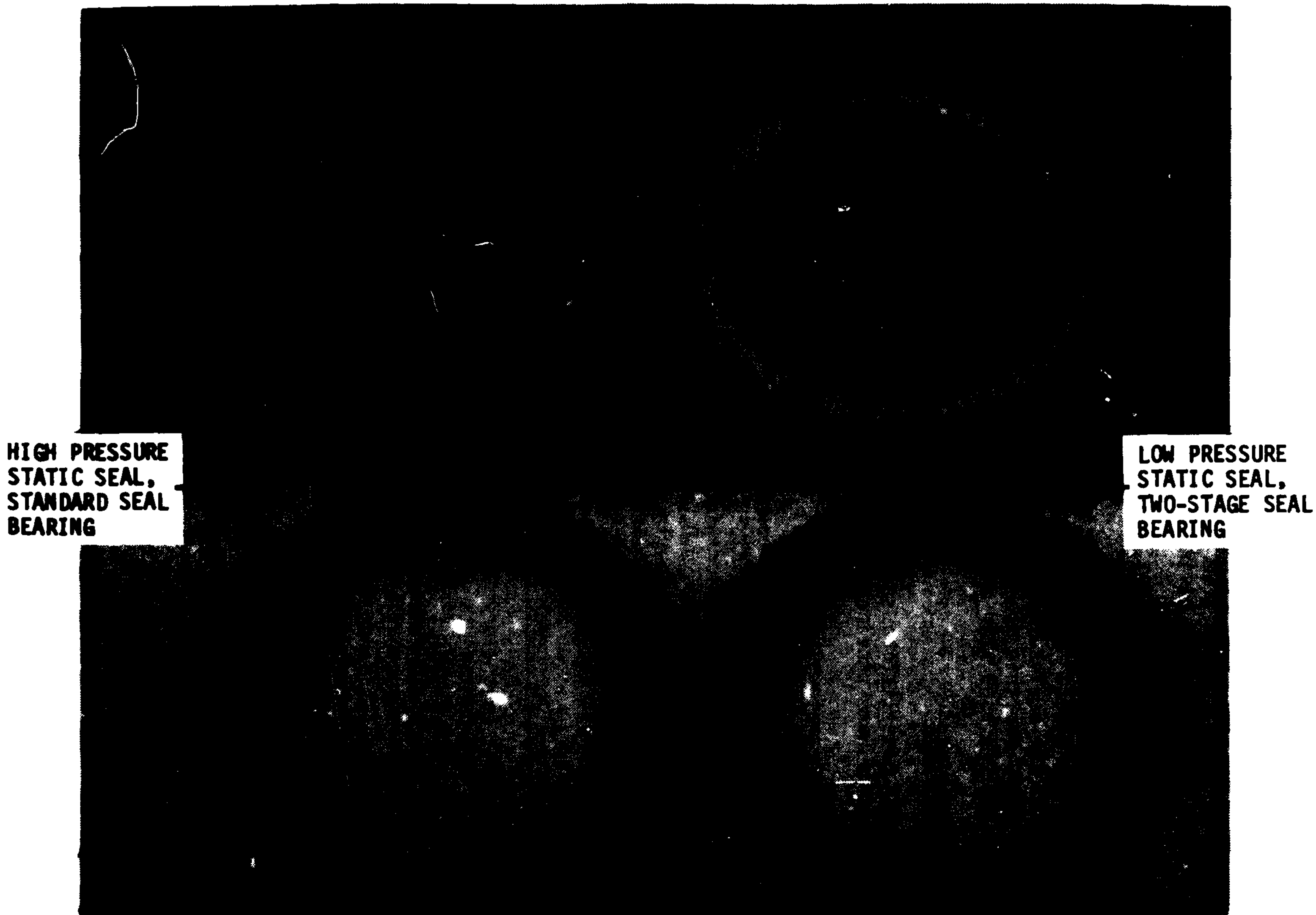


Figure 43. Military-standard external static seals (AO-8 fluid)

3.1.6.3.6 Military-Standard External Low-Pressure Static Seals

Examination of the AS568-218 size low-pressure static seal (Figure 43) from the B-52 seal bearing, revealed a thin cut which apparently was made when the bearing was removed from the end cap. The cut was all on the high-pressure side therefore having no effect on the seal's performance.

3.1.6.3.7 Military-Standard Internal High-Pressure Static Seals

On the larger AS568-330 size static seals (Figure 44), minimal deterioration was noted. Each O-ring had a radial cut in its side where the bias-cut end of the dual-turn backup ring was located. The backup ring seemed excessively short for this groove as there was approximately one-half inch underlap of the end bias when installed in the grooves.

3.1.6.3.8 Boss Seals

Metal boss fitting seals were used in the AO-8 test because O-rings of the wrong size were provided by the supplier.

3.1.6.3.9 Power-Transfer Cylinder

Upon disassembly of the power-transfer cylinder, some rust coloration on internal parts was discovered. The actuator piston (Figure 45) had a light reddish-brown discoloration on the rod side of the piston seal for approximately 120° of the piston O.D. Also, the discoloration was found in the test-fluid side of the piston-ring groove and on the end of the piston. The trunnion (Figure 46) had a dark reddish-brown discoloration in the test-fluid-hole relief slot. The piston leakage-hole relief slot, however, had no discernable discoloration.

3.1.6.3.10 Fluid and Component Discoloration Determinations

Chemical laboratory tests were run to identify the discoloration on the 4340-steel test components and the suspended material in the post-test fluid. A total acid number (TAN) was measured on the post-test fluid, and no change from the new fluid measurement of 0.047 mg. of KOH/gm was found. The dark material held in suspension in the post-test fluid sample was filtered and subjected to an infra-red spectroscopy and X-ray fluorescent scans. Iron was the only major element found, with neither phosphorus (from the seals) nor chlorine (from the fluid) detected. An infra-red spectroscopy scan was performed on the discolored material scraped from the 4340 steel test cylinder parts. The scrapings were identified as rust and a hydrocarbon.

No conclusion could be reached as to the origin of the hydrocarbon or the cause of the rust.

3.1.7 Seal-Test Conclusions

The Halocarbon AO-8 fluid/Firestone PNF seals performed as well as the MIL-H-5606 fluid/nitrile seals when compared by acceptable leakage levels. However, other secondary performance criteria had less than desirable results. These included the corrosion of the 4340-steel test cylinder, the adhesion of seal material in their glands and the particle generation by the seals.

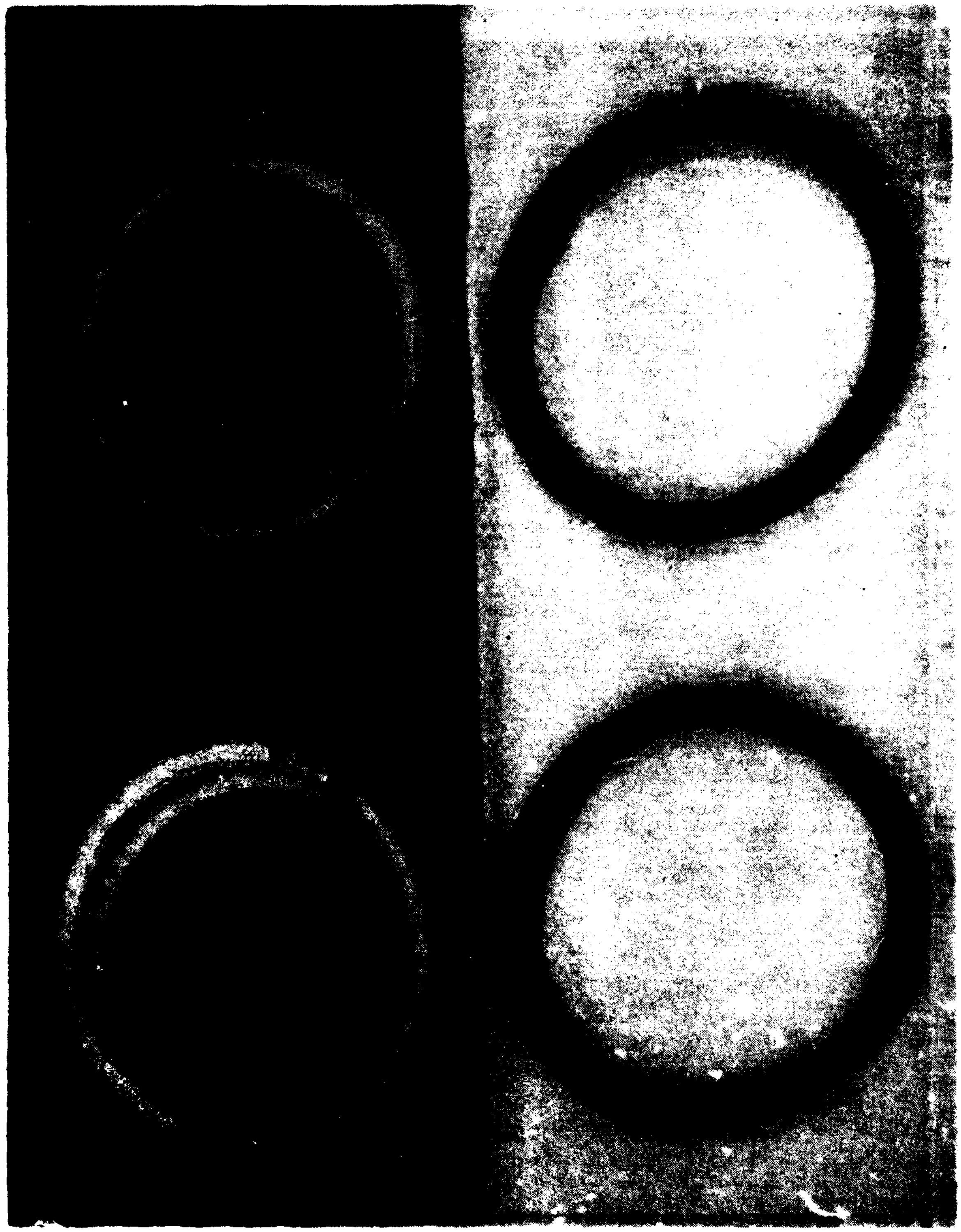


Figure 44. Military-standard internal static seals (AO-8 fluid)

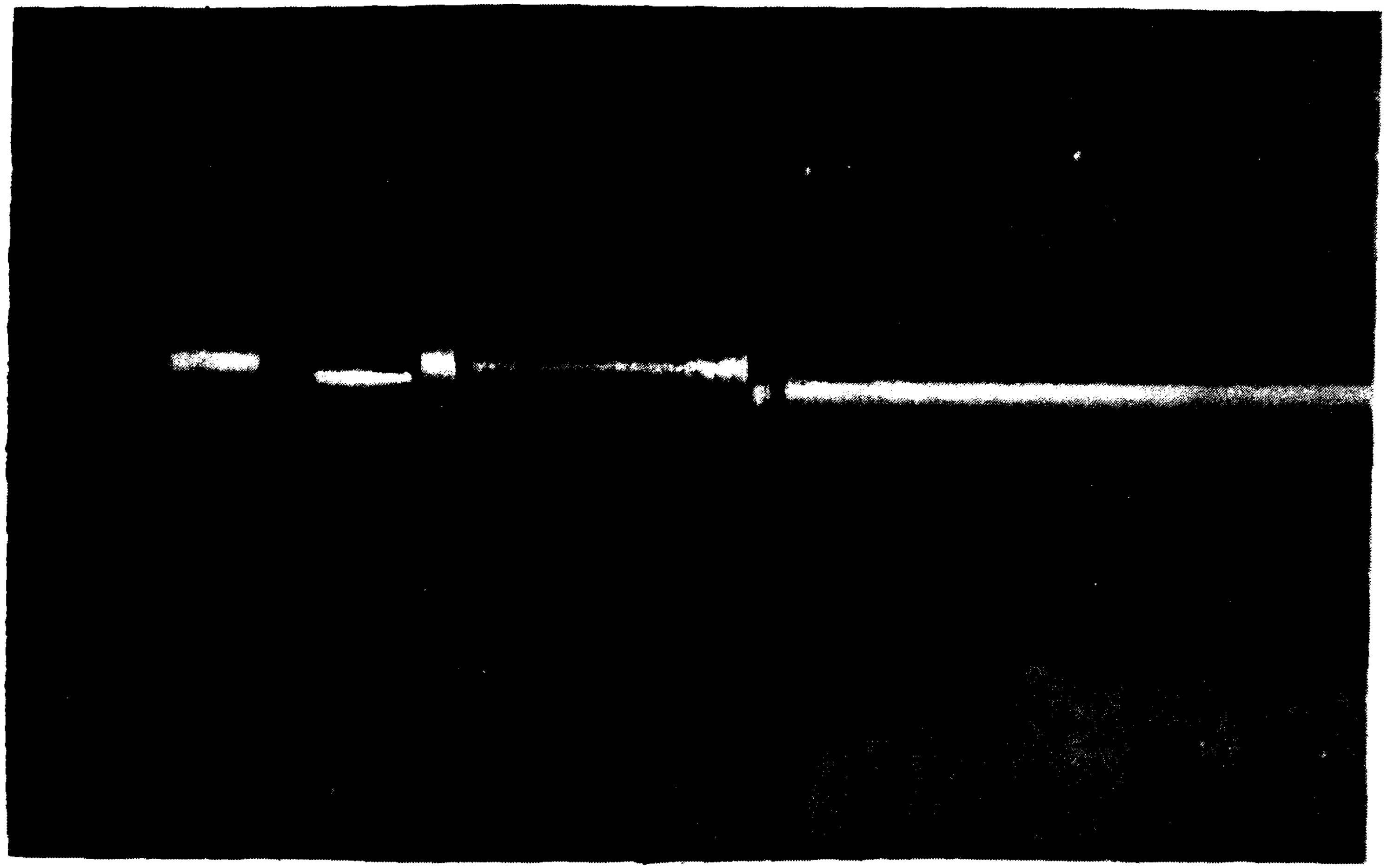


Figure 45. Power-transfer actuator piston after AO-8 fluid test

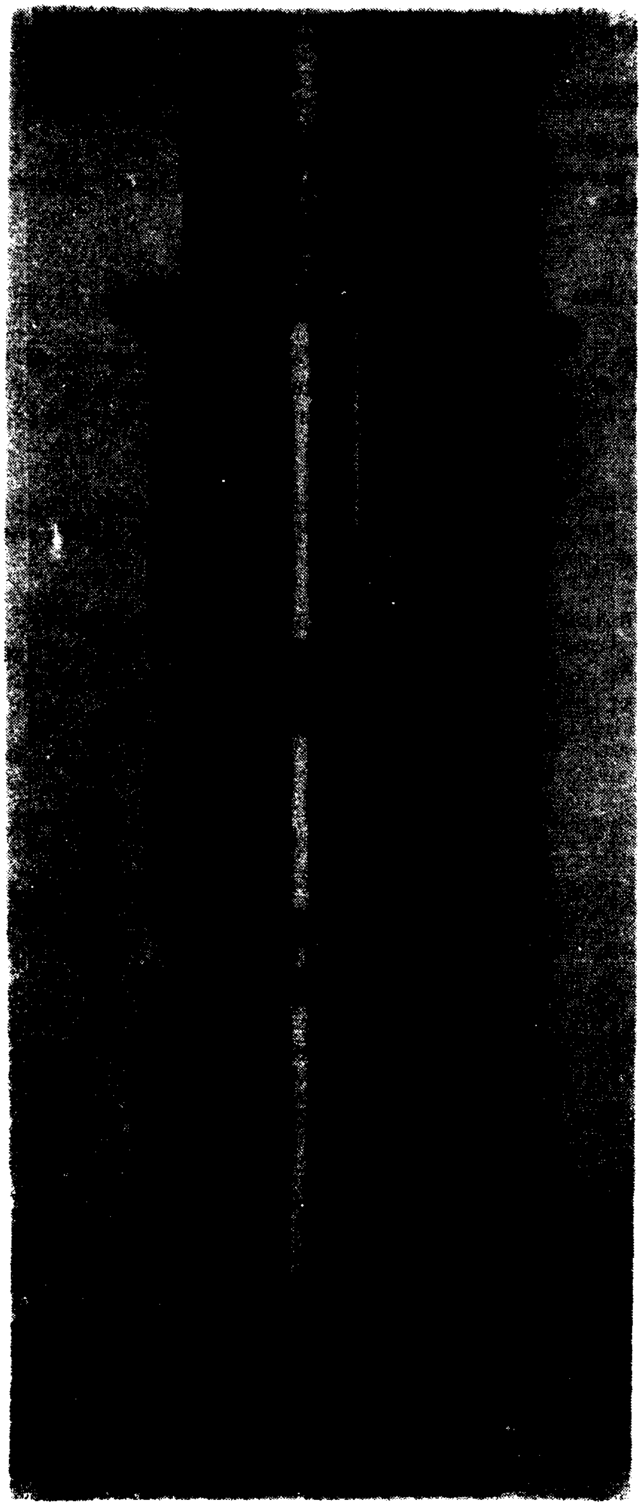


Figure 46. Power-transfer actuator trunnion after AO-8 fluid test

The virgin TFE backup rings had indications of approaching the useful upper temperature limit.

3.1.8 Seal-Test Recommendations

Following the review of the foregoing seal test results, the following recommendations were made relative to the additional component compatibility tests planned:

a. Boeing should continue into the 50-hour pump test with fluid as is.

b. The Aero Propulsion Laboratory and the Materials Laboratory should work with the Halocarbon Products Corporation to develop a rust inhibitor additive and determine the appropriate concentration required.

c. When a new fluid formulation is recommended by the Aero Propulsion Laboratory and the Materials Laboratory it should be utilized in the remainder of the component testing.

d. The Aero Propulsion Laboratory and the Materials Laboratory should work with the Firestone Tire and Rubber Co. to develop a lower swelling and tougher sufaced PNF material to be incorporated in the later phases of testing.

3.2 HYDRAULIC PUMP TESTS

The specified series of planned component compatibility tests included evaluations of a conventional hydraulic pump of the type used on present high performance aircraft to determine its compatibility with the AC-8 fluid from the standpoints of performance and operating life.

3.2.1 Fifty-Hour Pump Test

The first of these pump evaluations was a test for a minimum of fifty hours under various pump qualification test conditions to determine how the performance of the pump operating with AO-8 fluid compares with its operation with MIL-H-5606 fluid. A second goal was to determine the design changes or modifications required to obtain acceptable performance and endurance life as specified in MIL-P-19692.

For the fifty-hour pump test, the test fluid was Halocarbon AC-8, Batch No. 92978, which contained no antiwear additives. The elastomeric seals were molded from phosphonitrilic fluoroelastomer (PNF) Compound No. 200R-211658 supplied by The Firestone Tire & Rubber Company in Akron, Ohio. The O-rings were molded by Nichols Engineering, Inc. in Shelton, Connecticut, which has since been purchased by the Aerospace Products Division of the Lord Corporation. The special shaped elastomer seals for the pump housing and shaft seal were molded by the Precision Rubber Products Corporation in Lebanon, Tennessee.

3.2.1.1 Test Pump

In order to obtain a pump representative of current designs used on high performance aircraft for this test program, the leading manufacturers were contacted for their recommendations. The following models, which had been qualified to MIL-P-19692, were offered:

Abex Aerospace Division's Models:

AP10V-59	Displacement:	1.7 cubic inches per rev
	Rated flow:	26 gpm at 3750 rpm
AP12V-17	Displacement:	1.8 cipr
	Rated flow:	42.5 gpm at 5800 rpm

Sperry-Vickers Models:

PV3-022-2	Displacement:	0.22 cipr
	Rated flow:	9 gpm at 10,000 rpm
PV3-044-3	Displacement:	0.44 cipr
	Rated flow:	15 gpm at 8,000 rpm
PV3-075-1	Displacement:	0.75 cipr
	Rated flow:	21 gpm at 7,000 rpm

Due to the torque and power limitations of the laboratory varidrive units available for this test, the Sperry-Vickers PV3-075 size was the largest

of those offered which could be driven both at its rated conditions and at the overspeed and overload conditions required by MIL-P-19692C. Two Model PV3-075-15 pumps, which were in production for use on the F-16 emergency power unit (EPU), were offered and accepted.

The Model PV3-075-15 pump is one size in the Sperry-Vickers inline series of positive-displacement axial-piston pumps. It is a variable displacement unit with a nominal zero-flow delivery pressure of 3000 psi. Figure 47 is a cross-section view showing the drive shaft, supported by a radial bearing at each end, splined to the rotating cylinder block. The nine pistons, fitted into 0.448-inch diameter cylinder bores, are caused to reciprocate as their bearing shoes slide on the inclined plane of the shoe bearing plate which is supported by the yoke.

Piston stroke is controlled by the yoke angle which pivots on two trunnion bearings in response to pressure force on the actuator piston acting against the return spring. When the discharge pressure drops below the cutoff pressure (such as due to a system flow demand), the spring force pivots the yoke toward the maximum-stroke (full-flow) position; and, when the discharge pressure increases above the cutoff value (due to a reduction in flow demand), the actuator piston force pivots the yoke toward the minimum-stroke (zero-flow) position.

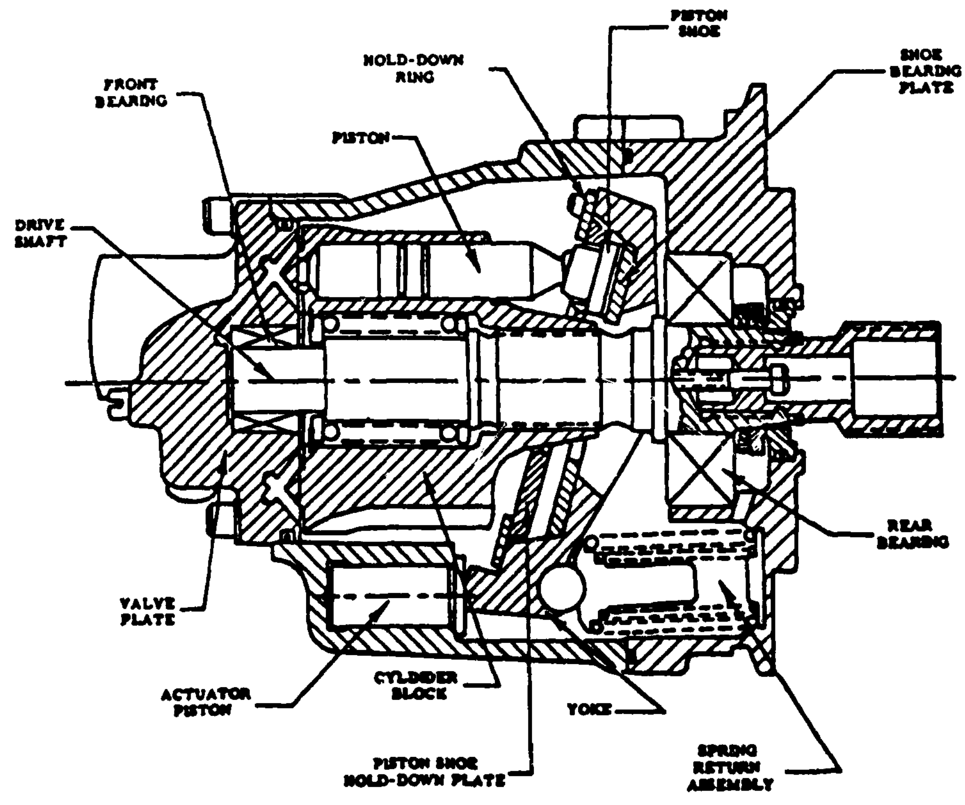


Figure 47 Pump cross section

The Model PV3-075-15 is nearly identical to, and was qualified by similarity to, the PV3-075-1 pump which was qualified to MIL-P-19692B. The primary differences are that, for the F-16 EPU, the rated speed was raised from 7,000 rpm to 7,500 rpm; and the following minor design changes were made to provide a minimum case drain flow of 0.5 gpm:

a. the size of the pressure-balancing holes in the piston shoes was reduced to reduce case leakage back through these holes during piston suction strokes.

b. The piston to cylinder bore clearance was increased .0003 inch to provide additional case leakage.

For the fire resistant aircraft hydraulic system program, the basic speed rating of 7,000 rpm was retained as the rating representative of most applications.

The pump used in the tests reported herein had previously been "run in" with hydraulic fluid per MIL-H-5606. Preparation of the pump for testing in the AO-8 fluid, included its disassembly, visual inspection of all parts, photographing of critical parts, measurement of certain dimensions, and the cleaning of metal parts per the procedure included in the Design Guide, Reference 1, Section 7.0. In addition, all elastomeric seals were replaced with seals molded from PNF compound as noted in 3.2.1 herein.

3.2.1.2 Test Setup

Testing was conducted in the Boeing Laboratory at Wichita, Kansas with the test setup as shown schematically in Figure 48. Pump inlet fluid was supplied by a nitrogen-pressurized free-surface non-separated reservoir. System volume pressurized by the pump was approximately 50 cubic inches; and, a 200 cubic inch aircraft accumulator with the piston removed was added to simulate a more representative system compliance volume for the maximum pressure, response time, and pressure pulsation tests. Two fast-acting solenoid valves with variable-restricting throttle valves in series were used to control discharge flow for various test conditions including the endurance tests.

The complete equipment list is as follows:

1. Pump, Sperry-Vickers PV3-075-15, S/N MX-319687
2. Varidrive, U.S. Electric Motors VEU-GSDT, S/N 940865
3. Torque Transducer, Lebow Associates 1235-107, S/N 121
4. Pressure Filter, Aircraft Porous Media AD-3255-16Y77
5. Discharge Pressure Transducer, CEC 4-326-008, S/N 17413
6. Discharge Pressure Gage, Classco 0-5000 psi 1/4% accrcy, S/N RL4086
7. Discharge Flowmeter, Cox AN8-4, S/N 22013
8. Relief Valve, Droitcour Co. AN6279-12
9. Solenoid Valve, Marotta 280203, S/N 106
10. Solenoid Valve, Marotta 280203, S/N 107
11. Load Valve, Parker Hannifin MV-1230-S
12. Compliance Volume Accum., Parker Aircraft 2660359, 200 cu. in., S/N A101
13. Compliance Volume Valve, Parker Hannifin MV-1230-S
14. Case Drain Pressure Transducer, CEC 4-326, S/N 17907

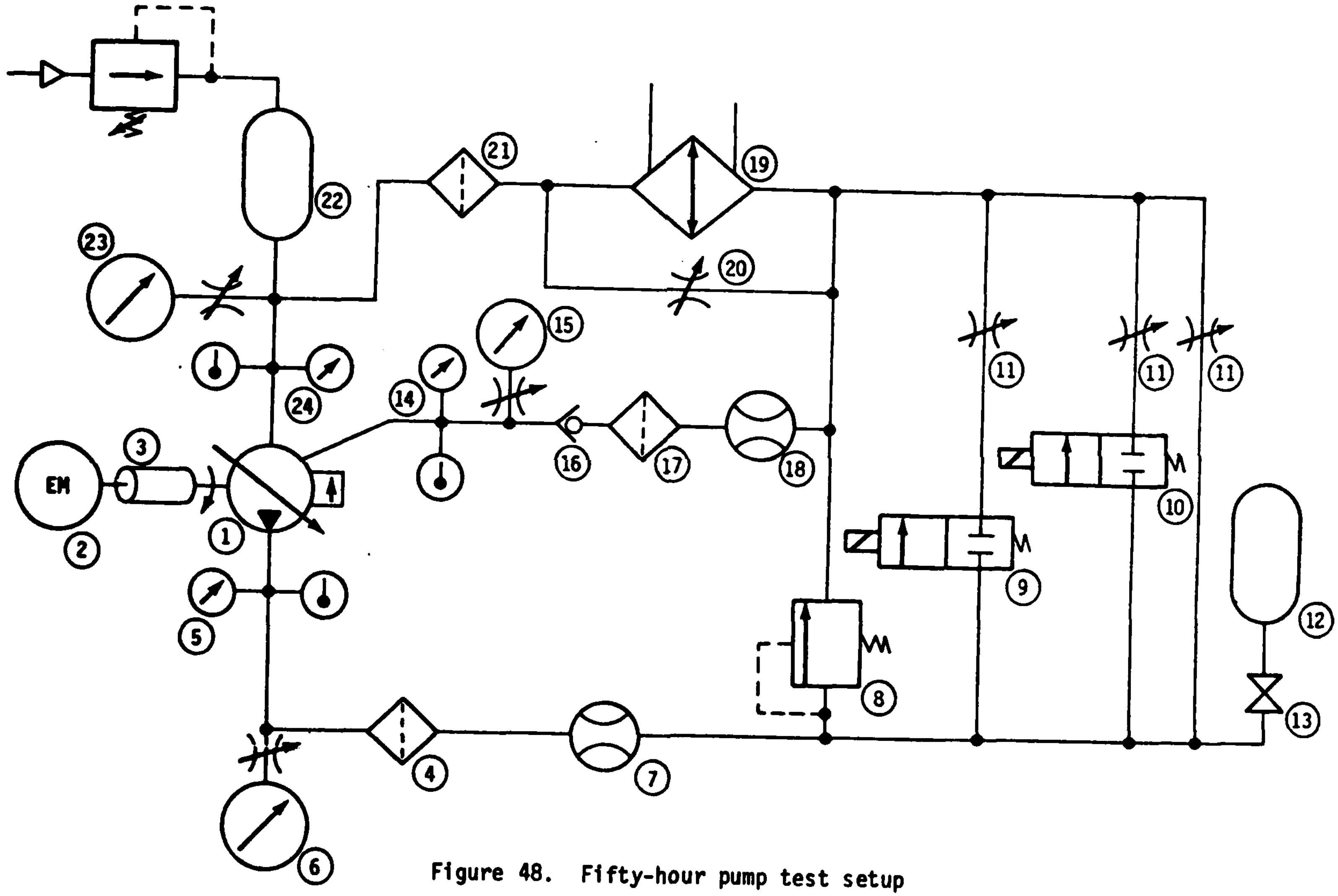


Figure 48. Fifty-hour pump test setup

15. Case Drain Pressure Gauge, Ashcroft 0-300 psi S/N RL4059
16. Check Valve, Republic Mfgr. AN6280-8
17. Case Drain Filter, Aircraft Porous Media AC-3258-68Y15
18. Case Drain Flowmeter, Cox AN8-4, S/N 11672
19. Heat Exchanger, Sperry-Vickers
20. Bypass Valve, Parker Hannifin MV-1230-S
21. Return Filter, Aircraft Porous Media AD-3255-16Y77
22. Reservoir, Boeing Co. 35-3297
23. Inlet Pressure Gage, Roylyn Lindsay 300 psi
24. Inlet Pressure Transducer, CEC 4-326-001, S/N 21942

3.2.1.3 Test Procedure

Testing was conducted in accordance with the test plan shown in Appendix C except for deviations necessitated by shortcomings in the test circuit and the instrumentation. The following individual test operations were performed:

3.2.1.3.1 Performance Tests

A series of calibration tests were run at constant speed and inlet fluid temperature conditions to determine input torque, outlet pressure, and pump efficiency for a number of flow conditions between zero flow (at rated discharge pressure) and maximum flow at maximum full-flow pressure. In addition, at each of the constant speed and inlet temperature conditions, a number of runs were made to determine the delivery flow, case drain flow, and pump efficiency for discharge pressures between 2500 and 1500 psi. The constant speed and inlet fluid temperature conditions were as follows:

a. 50% Rated Speed: 3,500 rpm, 180F and 240F
b. 75% Rated Speed: 5,250 rpm, 185F and 240F
c. Full Rated Speed: 7,000 rpm, 190F and 240F

For each of those conditions, the varidrive was set at the desired speed, the inlet temperature stabilized, and the discharge flow adjusted in several increments between the zero-flow rated discharge pressure condition and the maximum full-flow pressure condition. Then the pressure was reduced, in several increments, from 2500 psi to 1500 psi. At each point, the following data were recorded:

a. Pump speed
b. Input torque
c. Delivery flow
d. Case drain flow
e. Inlet pressure
f. Discharge pressure
g. Case drain pressure
h. Inlet fluid temperature
i. Discharge temperature
j. Case drain temperature

3.2.1.3.2 Critical Inlet Pressure Tests

This series of tests was run to determine the minimum inlet pressures required to obtain full delivery flow at various speeds and inlet fluid temperatures. The tests were run at the following constant speed and inlet fluid temperature conditions:

a. 50% Rated Speed: 3,500 rpm, 120F, 180F, and 240F
b. 75% Rated Speed: 5,250 rpm, 120F, 180F, and 240F
c. Full Rated Speed: 7,000 rpm, 120F, 180F, and 240F

For each of those conditions, the varidrive was set at the desired speed, the inlet temperature stabilized, the discharge pressure adjusted to the maximum full-flow pressure and maintained at that setting, and the pump inlet pressure decreased in 2 psi increments starting at approximately 60 psia. At each point, the following data were recorded; and, the inlet pressure reductions continued until the discharge flow had decreased a significant amount of the normal full-flow value:

a. Pump speed
b. Inlet fluid temperature
c. Discharge pressure
d. Inlet pressure
e. Delivery flow

3.2.1.3.3 Maximum-Pressure, Response-Time, and Pressure-Pulsation Tests

These tests were run to determine if the applicable requirements of MIL-P-19692C (which were met with a Model PV3-075-1 pump operating with fluid per MIL-H-5606) can be met with the test pump operating with AO-8 fluid. For these tests, the valve to the 200 cubic inch accumulator shell was alternately opened to provide a degree of system impedance which provides a rate of pump discharge pressure rise representative of an actual aircraft system, and closed to measure the higher pressure which would be obtained in a small close-coupled system.

3.2.1.3.3.1 Maximum-Pressure Tests

Transient pressure peaks obtained by suddenly reducing the pump flow demand from full flow at maximum full-flow pressure to zero flow at rated steady-state discharge pressure were determined at the following speeds at inlet fluid temperatures of 180F and above:

a. 50% Rated Speed: 3,500 rpm, 180F and 215F
b. Full Rated Speed: 7,000 rpm, 229F and 240F

For each of those conditions, the varidrive was set at the desired speed, the inlet temperature stabilized, and the discharge pressure adjusted to the maximum full-flow pressure. Then, the fast-acting solenoid shutoff valve upstream of the flow-controlling throttle valve was energized to the shutoff position. At each point, the discharge pressure was recorded on a high-speed oscillograph and the tape retained for the record.

3.2.1.3.3.2 Response-Time Tests

Response times as defined in MIL-P-19692B and C, were determined during rapid changes from high-flow to low-flow and from low-flow to high-flow conditions at the following speeds and nominal inlet fluid temperatures:

a. 50% Rated Speed: 3,500 rpm, 180F and 215F
b. 75% Rated Speed: 5,250 rpm, 190F and 220F
c. Full Rated Speed: 7,000 rpm, 220F and 220F

For each of those conditions, the varidrive was set at the desired speed and the two flow-controlling throttle valves adjusted so that, with flow through both valves, the pump delivery flow was approximately 90% of full flow; and, so that, with flow through the most restricted valve, the pump delivery flow was approximately 5% of full flow. With the inlet temperature stabilized, and both flow-controlling throttle valves opened for the 90% delivery flow, the high-speed oscillograph was started, the fast-acting solenoid valve upstream of the least restricted flow controlling valve was shutoff to reduce delivery flow to the 5% value, and the discharge pressure transient was recorded. Then, the procedure was repeated by opening the closed solenoid valve; and, the discharge pressure transient caused by increasing delivery flow from the 5% value to 90% value was recorded. All oscillograph tapes were retained for the record.

3.2.1.3.3.3 Pressure-Pulsation Tests

The oscillograph records taken during the foregoing tests were examined and the maxinum steady-state peak-to-peak pulsation pressures were measured.

3.2.1.3.4 Endurance Tests

The remaining 16.25 hours of the Fifty-Hour Pump Test were divided between two normal-rated-load test phases and an overload test phase per MIL-P-19692C as follows and as shown in Table 22.

3.2.1.3.4.1 Normal-Rated-Load Endurance Tests

Five hours of testing were run at conditions approximating those specified in MIL-P-19692C, Table IV, Phase 6. The flow-controlling throttle valves were adjusted so that with the varidrive set to maintain the pump at its rated speed: 7,000 rpm, the delivery flow could be alternated between 70% and 5% of rated delivery flow. Then with the pump inlet pressure at approximately 90 psig and inlet fluid temperature averaging 190F, the pump was run 5.0 hours at 7,000 rpm with the flow alternating between 70% of rated delivery flow for 5 seconds and 5% of rated delivery flow for 5 seconds.

An additional 4.5 hours were run at conditions approximating those specified in MIL-P-19692C, Table IV, Phase 7. With the pump inlet pressure at approximately 90 psig, and the inlet fluid temperature averaging 204F, the pump was run at 7,000 rpm with the flow alteranting between 5 seconds at 100% rated delivery flow and 5 seconds at zero flow.

3.2.1.3.4.2 Overload Endurance Test

Six and three quarters hours of testing were run at conditions approximating those specified in MIL-P-19692C, Table V, Phase 7, except that the 125% of rated speed was reached in five incremental steps. The pump compensator valve was adjusted, with its adjusting screw and locknut, to produce a discharge pressure of 3,750 psi (125% of rated discharge pressure) at zero delivery flow; and, one of the flow-controlling throttle valves was adjusted for each of the five incremental speeds so that the pump discharge pressure was approximately 3,300 psi (110% of rated discharge pressure). The pump was run at each of the speed and inlet fluid conditions, for the duration

TABLE 22 FIFTY-HOUR PUMP ENDURANCE TEST LOG

(NORMAL AND OVERLOAD ENDURANCE TEST PER MIL-P-19692C)

DATE	PUMP RPM	PRESSURE PSIG IN	PRESSURE PSIG OUT	PRESSURE PSIG CASE	TEMPERATURE °F (AVERAGE) IN	TEMPERATURE °F (AVERAGE) OUT	TEMPERATURE °F (AVERAGE) CASE	HRS.	HRS. THIS PHASE	PHASE	TABLE
4-7-79	7000	88/85	2975/2925	109/96	190	200	238		5.0	6	IV
4-9-79	7000	91/86	2950/2850	122/100	204	213	250		4.5	7	IV
4-11-79	7700	123/116	3775/3350	160/130	214	225	258	2.0		7	V
4-11-79	8000	108/115	3775/3300	154/124	238	247	285	0.5		7	V
4-11-79	8200	108/115	3725/3250	124/154	238	248	287	0.5		7	V
4-11-79	8400	104/97	3750/3250	148/118	232	243	282	1.5		7	V
4-12-79	8750	105/98	3750/3250	150/120	238	249	287	2.25	6.75	7	V

indicated in Table 22 with the delivery flow alternating between full flow for 5 seconds and zero flow for 5 seconds.

3.2.1.3.5 Post-Test Inspections

Following the completion of the normal-rated-load endurance tests, and again following completion of the overload endurance test, the pump was disassembled and visually and dimensionally inspected for deterioration, wear, and evidence of fluid breakdown.

3.2.1.4 Test Results

The data taken and a discussion of the results for each test, including comparisons with data taken during the qualification of the Model PV3-075-1 pump with petroleum base fluid per MIL-H-5606, are as follows. The aforementioned qualification data was found in Reference 8; and, the pertinent data used in the following comparisons is included herein in Appendix D.

Initially, it was planned to run a number of the tests with three different inlet fluid temperature conditions: 120F, 180F, and 240F. However, it was found that the system heat exchanger had insufficient cooling capacity to hold the inlet fluid temperature to 120F. It was also found that the torque meter brushes suffered early failure at the 240F inlet temperature. Therefore, complete data were taken only at the intermediate inlet fluid temperature: approximately 180F.

3.2.1.4.1 Steady-State Performance

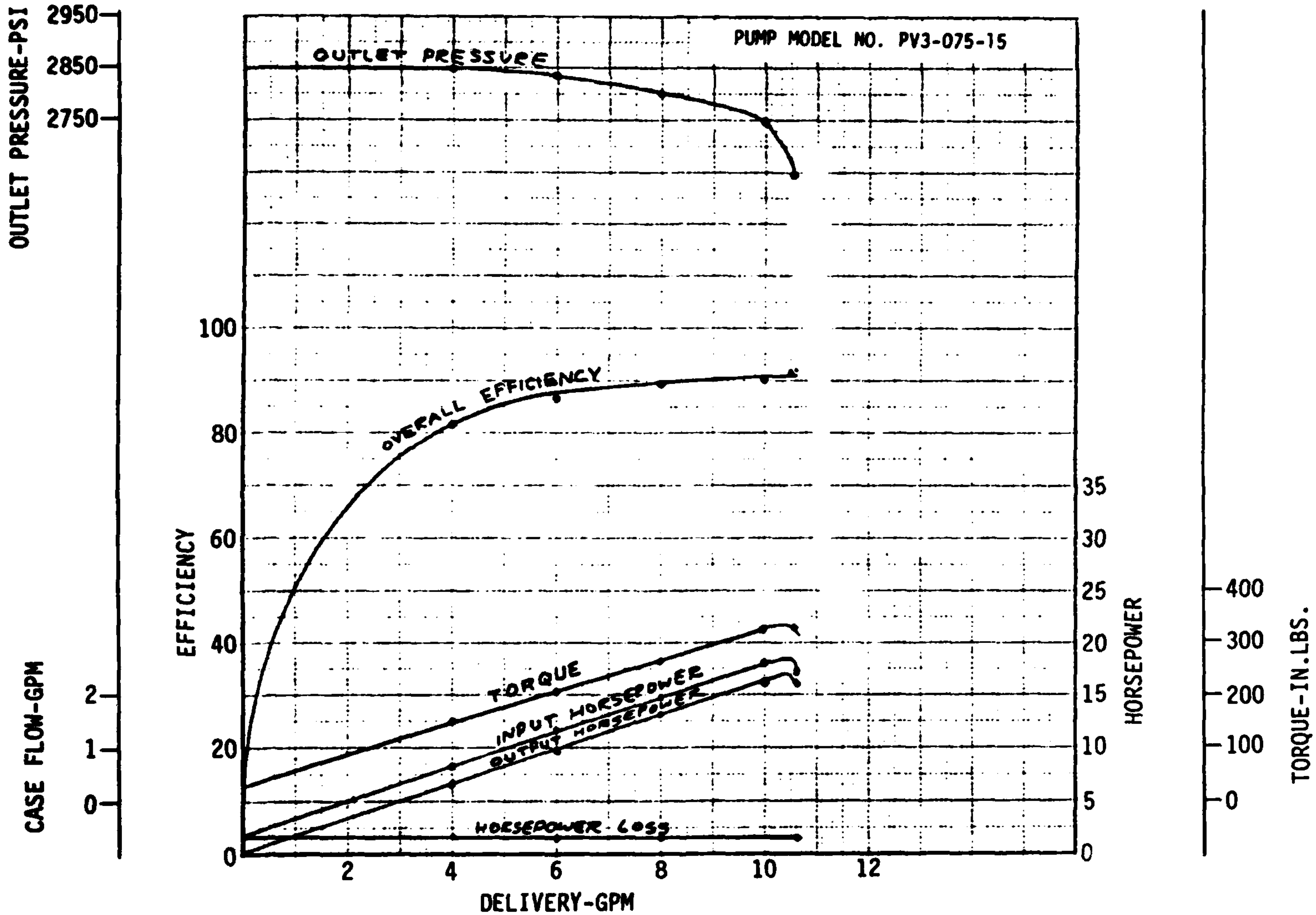
The data for each of the three speed conditions: 3,500, 5,250, and 7,000 rpm, are shown in the curves in Figures 49 through 54. In Figures 49, 51, and 53, the torque and outlet pressures measured at various delivery flow conditions between zero flow and full flow are shown along with the calculated values for input and output power, power loss, and overall efficiency. In Figures 50, 52, and 54, delivery flow rate and case drain flow rate at various full-flow pressure conditions are shown along with the calculated values of volumetric efficiency and overall efficiency.

The steady-state performance of the PV3-075-15 pump operating with AO-8 fluid at 180-190F inlet temperature was almost identical to the performance of the PV3-075-1 pump operating with MIL-H-5606 fluid at 240F inlet temperature. The following relationships can be seen by comparing the data for the "AO-8 pump", displayed in Figures 49 through 54, with the data for the "5606 pump" displayed in Figures D-1 through D-6 in Appendix D:

a. For each of the three speed conditions, the delivery flow and case drain flow from the two pumps were very nearly equal at all points.

b. The input torque, input power, and power output for the AO-8 pump were lower than that for the 5606 pump; however, those differences were primarily due to the fact that the AO-8 pump was run at a lower discharge pressure (2950 psi at zero flow compared to 3125 psi for the 5606 pump).

8. Sperry Vickers Report, Qualification Test of Pump Model PV3-075-1 to MIL-P-19692B, Project No. 8-0115-204-340, February 29, 1968.



INLET TEMP: 180 ± 20F

FLUID: HALOCARBON AO-8

Figure 49. Fifty-hour pump partial-flow performance at 3500 rpm

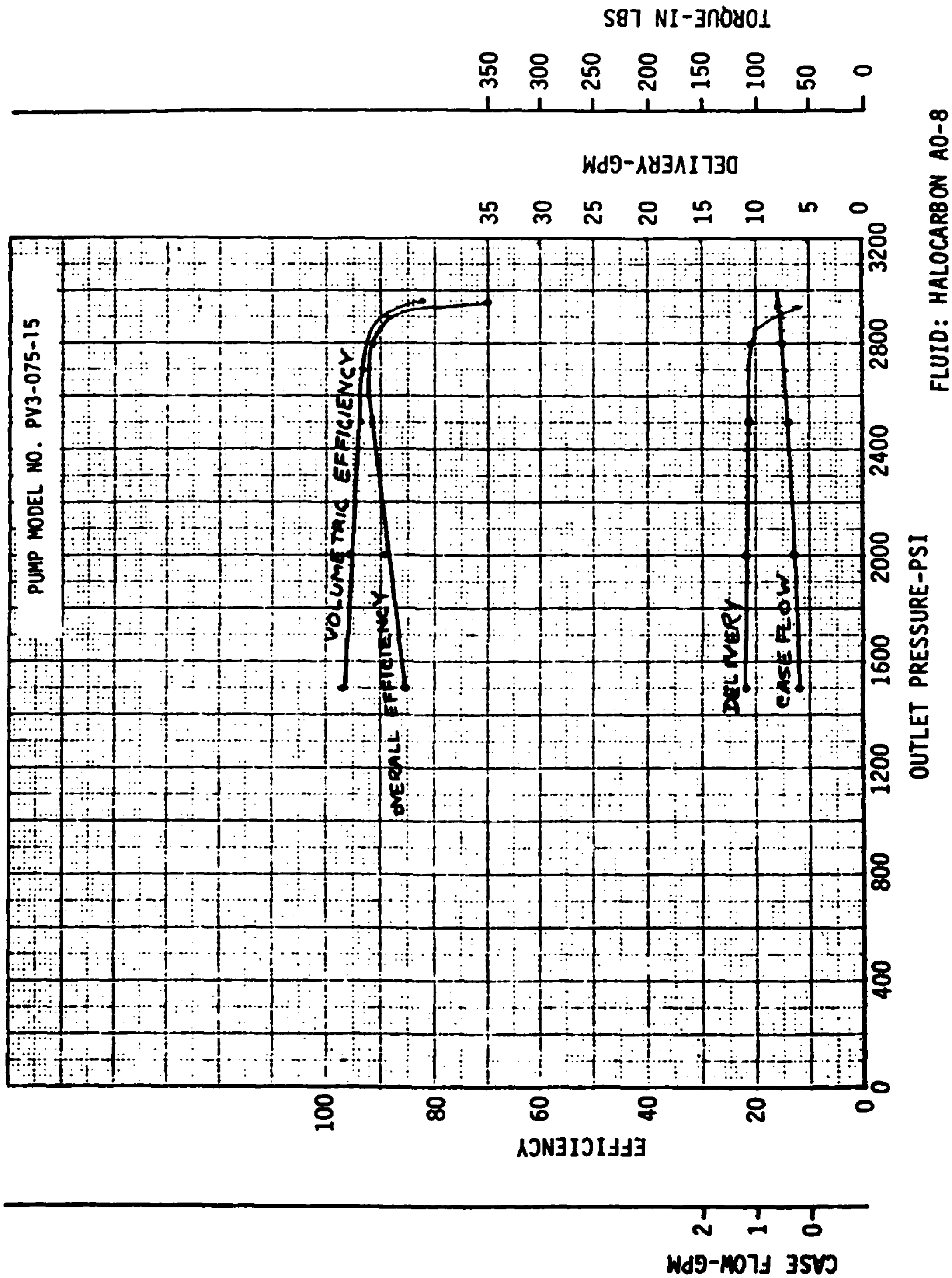
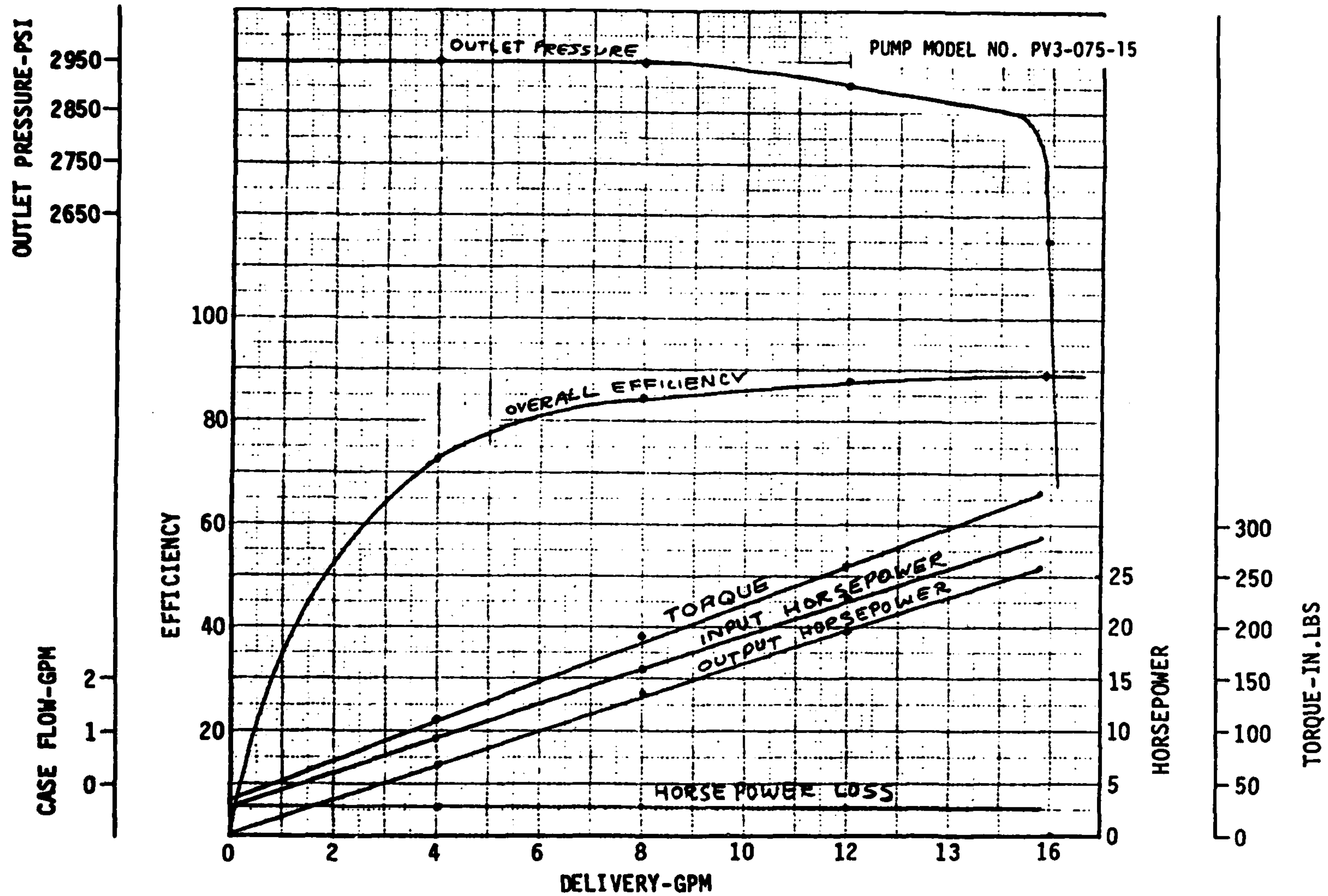


Figure 50. Fifty-hour pump full-flow performance at 3500 rpm



INLET TEMP: 185 ±15F FLUID: HALOCARBON AO-8

Figure 51. Fifty-hour pump partial-flow performance at 5250 rpm

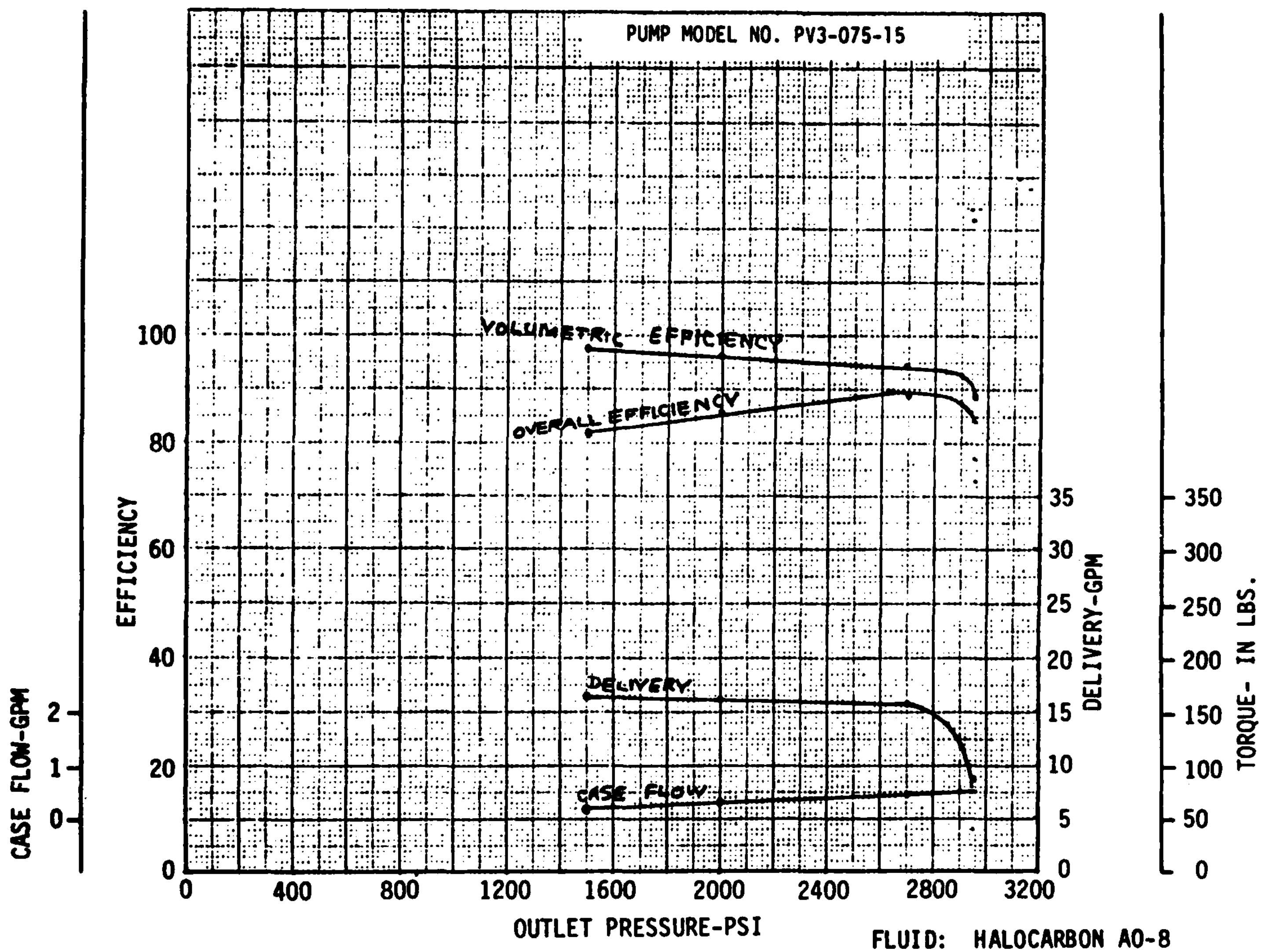
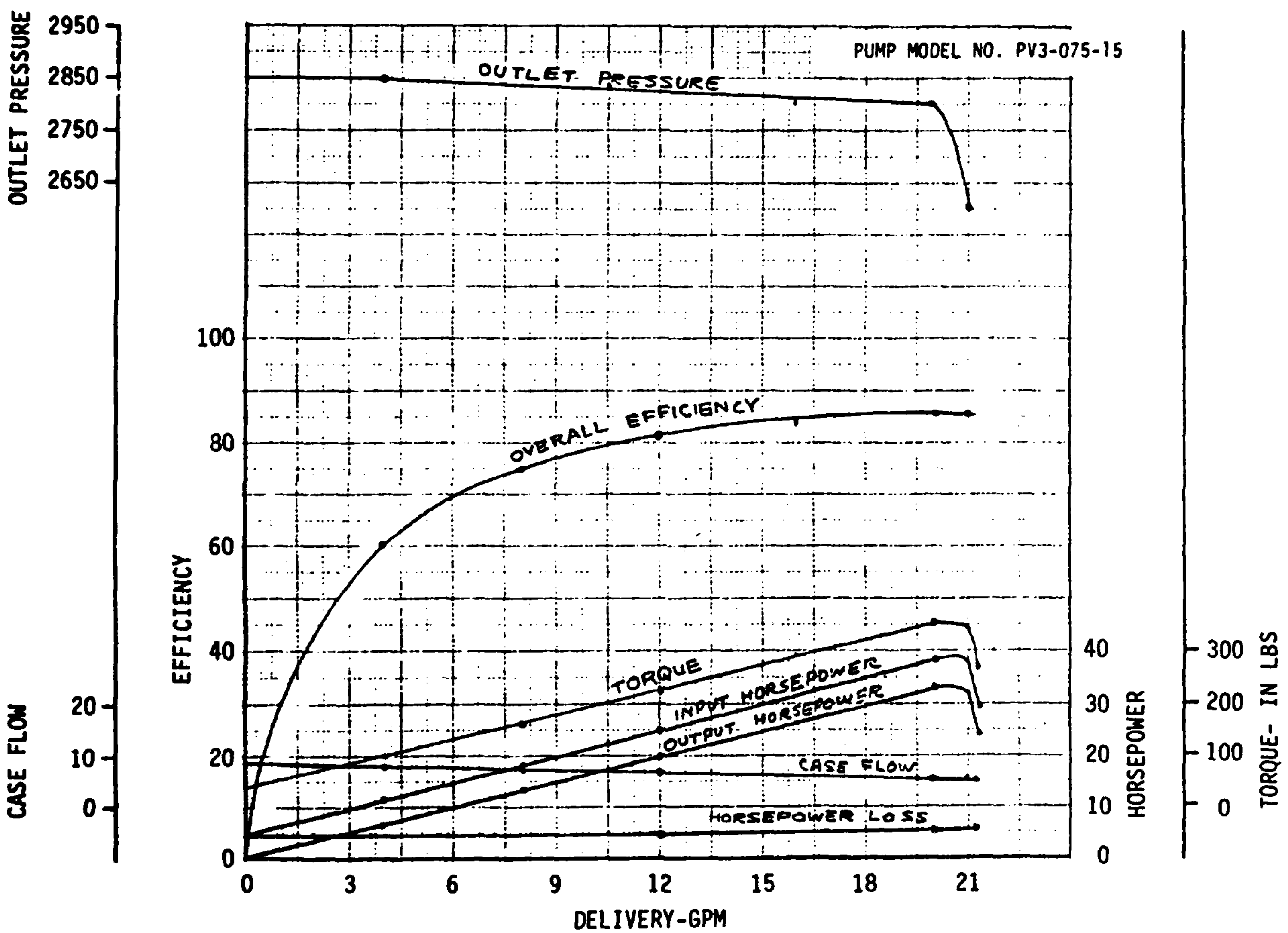


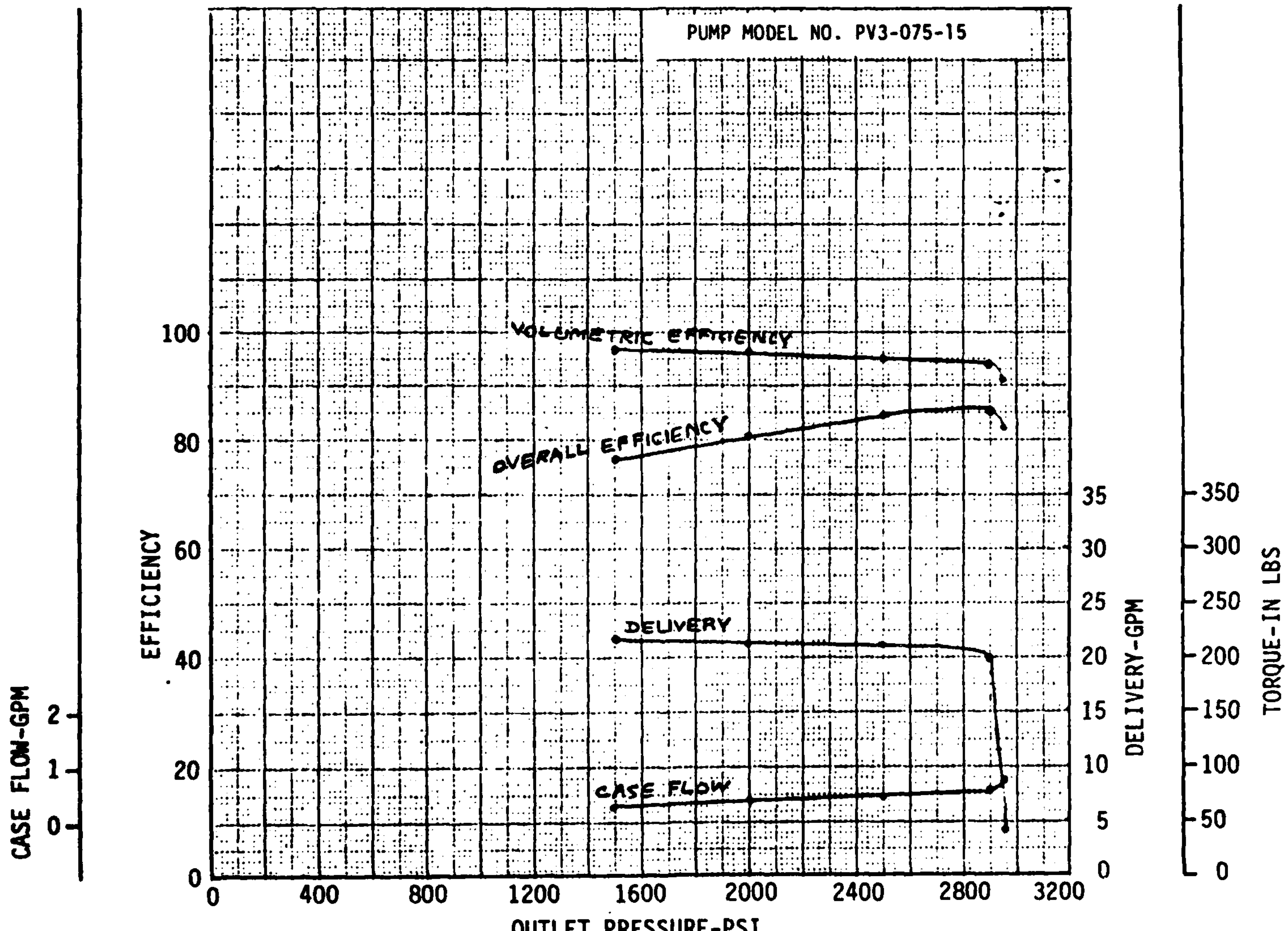
Figure 52. Fifty-hour pump full-flow performance at 5250 rpm



INLET TEMP: 190 ± 10F

FLUID: HALOCARBON AO-8

Figure 53. Fifty-hour pump partial flow performance at 7000 rpm



INLET TEMP: 190 ± 10F

FLUID: HALOCARBON AO-8

Figure 54. Fifty-hour pump full-flow performance at 7000 rpm

c. At the 3500-rpm speed condition (50% of rated speed), the AO-8 pump had lower power loss and higher overall efficiency in the cutout flow regime (zero flow to full flow at full-flow pressure). As seen in Figure 49, the peak efficiency for the AO-8 pump with inlet fluid temperature of 180F was 91.51% compared to 86% for the 5606 pump with inlet fluid temperature of 240F (see Figure D-2). In the full-flow regime at lower pressures, the volumetric efficiencies for the two pumps were nearly identical at each respective data point, but the peak overall efficiency for the AO-8 pump was slightly higher, eg. 92% comapred to 90%.

d. At the 5250-rpm speed condition (75% of rated speed), the power loss of the AO-8 pump was only slightly lower; and, the overall efficiency (89.5% max.) was only slightly higher than for the 5606 pump (see Figures 51 and D-4).

e. At the 7000-rpm speed condition (rated speed), the power losses and efficiencies of the two pumps were nearly identical (86% max., Figures 53 and D-6).

From these comparisons, it was concluded that no redesign was required to obtain rated delivery flow, with acceptable power input levels, from the PV3-075 pump with AO-8 fluid. However, it should be noted that higher inlet pressures are required. This is discussed in the following section.

3.2.1.4.2 Critical Inlet Pressures

The data for each of the three inlet fluid temperature conditions at each of the three pump speed conditions, 3,500, 5,250, and 7,000 rpm, are shown in the curves in Figure 55. In each case, the delivery flow decreased gradually as inlet pressure was reduced until a pressure was reached where a sharp dropoff in delivery, indicating the onset of cavitation, occurred. The following tabulation indicates the approximate critical inlet pressure for each speed and inlet fluid temperature condition as selected from the curves in Figure 55 as the pressures where a five percent drop in the maximum measured delivery flow occurred.

Pump Speed	Critical Inlet Pressure (psia)		
(rpm)	@ 120F	@ 180F	@ 240F
3,500	22.5	24.5	22.5
5.250	33.5	31.5	32.5
7,000	52.5	53.5	58.5

By comparison, the critical inlet pressure measured with the PV3-075-1 pump operating with MIL-H-5606 fluid at 7,000 rpm and 240F inlet fluid temperature was 24.5 psia. See Figure D-7 in Appendix D.

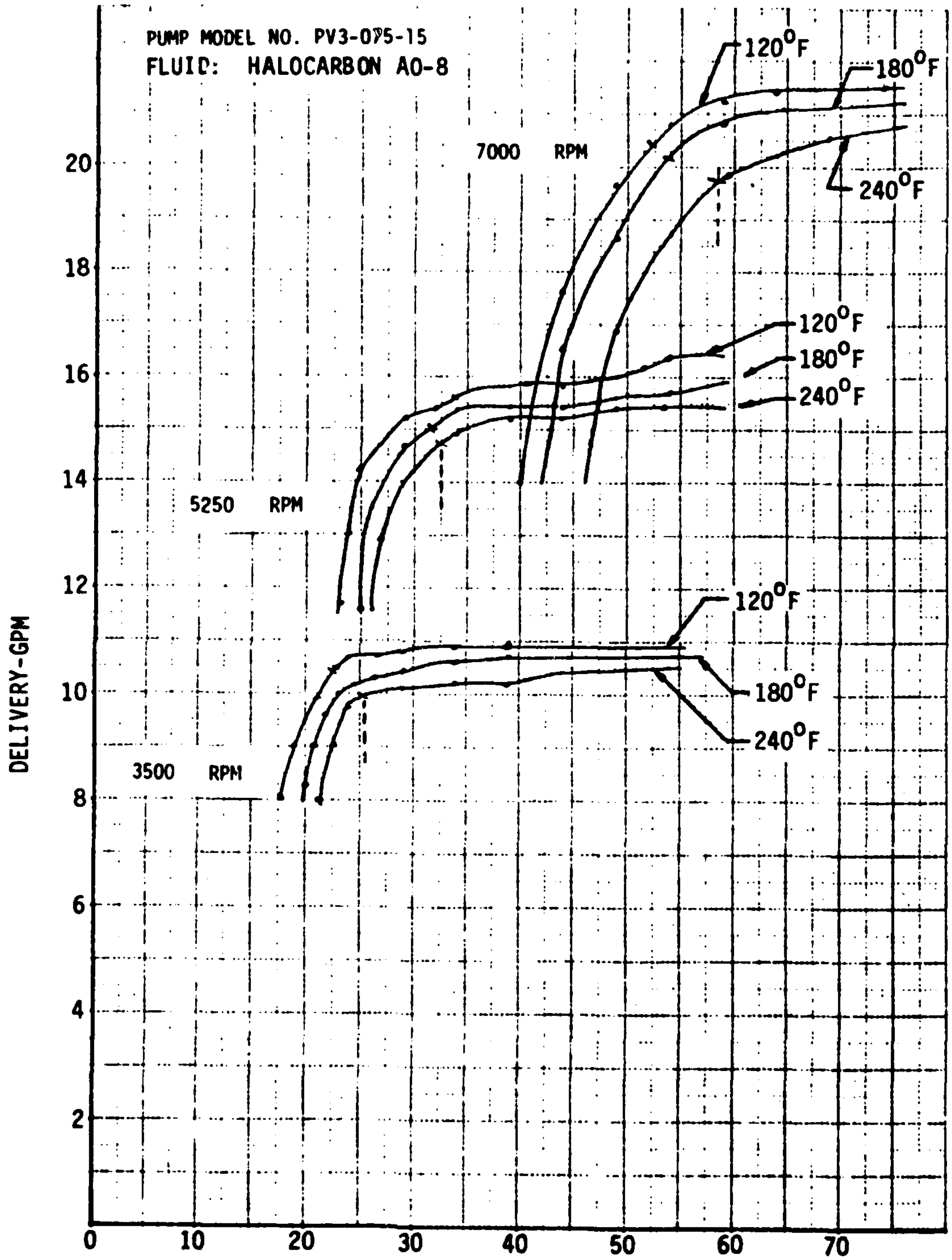


Figure 55. Fifty-hour pump critical inlet pressures

3.2.1.4.3 Maximum-Pressure, Response-Time, and Pressure-Pulsation Measurements

As noted in the test procedure, Section 3.2.1.3.3, these tests were run both with the basic system compliance volume (approximately 50 cubic inches) and with the 200 cubic inch accumulator shell added (250 cu in total).

3.2.1.4.3.1 Maximum Pressures Due to Sudden Valve Closing

The oscillograph traces of discharge pressure taken at the two speed conditions, with and without the added system compliance volume, are shown in Figures 56 and 57. As shown in the following tabulation, the peak pressure transient did not exceed the 135-percent of rated discharge pressure limit specifiedin MIL-P-19692C except during the 7,000-rpm speed condition with just the basic (50 cu in) compliance volume.

Pump Speed (rpm)	Transient Press. Peak (250 cu in Comp. Vol)			Transient Press. Peak (50 cu in Comp. Vol)		
	Temp.	(psi)	(%2950)	Temp.	(psi)	(%2950)
3,500	180F	3,550	120%	215F	3,850	130%
7,000	229F	3,700	125%	240F	4,225	143%

By comparison, the maximum pressures measured with the PV3-075-1 pump operating with MIL-H-5606 fluid in a similar test with a blocking valve directly at the pump outlet were as follows (see Figures D-10 and D-11):

Pump Speed (rpm)	Inlet Temp. (Deg. F)	Transient Press. Peak (psi)	(%3150)
3,500	220	3,700	117
7,000	240	3,950	125

From these comparisons, it is apparent that higher transient peak pressures are to be expected with the AO-8 fluid but that they will probably remain within the 135% limit except in very close-coupled systems with little compliance volume.

3.2.1.4.3.2 Response-Time Measurements

The oscillograph traces of discharge pressure vs time taken at the three speed conditions, for both a sudden valve closing and a sudden valve opening, with and without the added 200 cubic inch compliance volume are shown in Figures 58 through 63. The measured response times, with and without the added compliance volume, are tabulated in Tables 23 and 24 respectively. The response times measured during the qualification test of the PV3-075-1 pump, operating with fluid per MIL-H-5606 in a system with very little compliance volume (8-10 cu. in.), are tabulated in Figure D-8 in Appendix D.

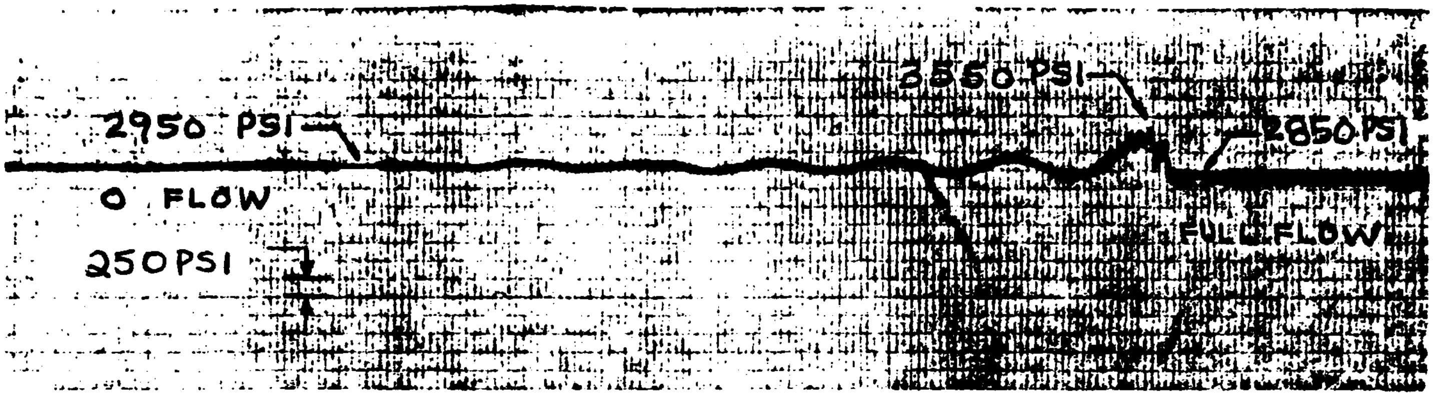


Figure 56a. Pressure transient due to a sudden valve closure while pumping AO-8 fluid into a 250-cu.in. system

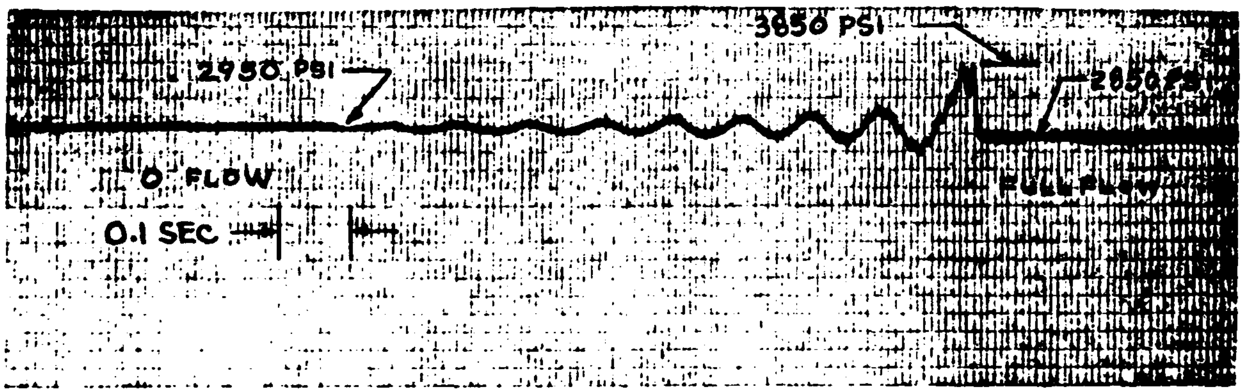


Figure 56b. Pressure transient due to a sudden valve closure while pumping AO-8 fluid into a 50-cu.in. system

Figure 56. Fifty-hour pump transient pressures at 3500 rpm

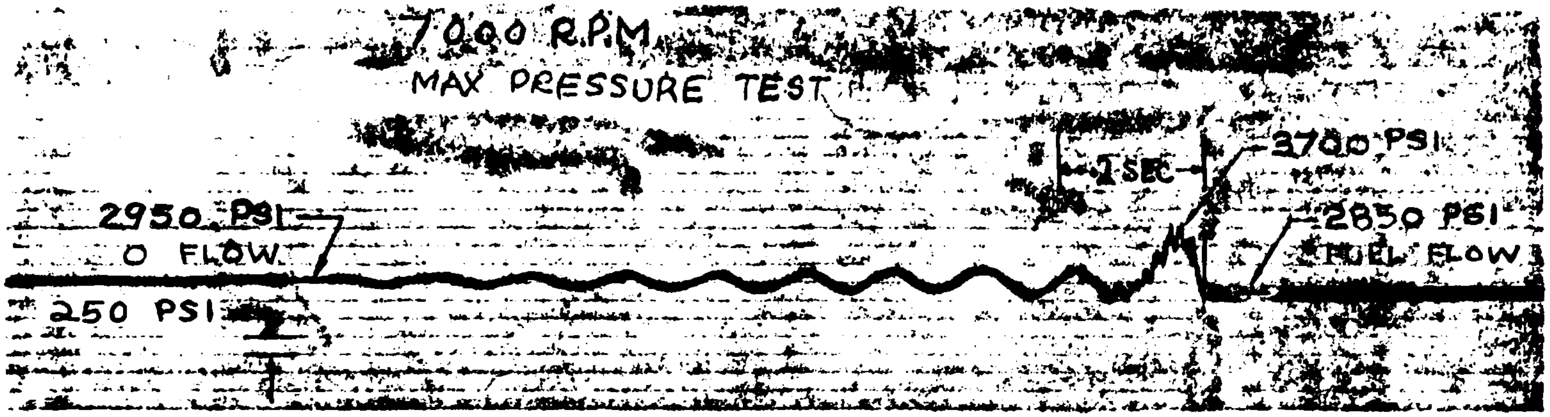


Figure 57a. Pressure transient due to a sudden valve closure while pumping AO-8 fluid into a 250-cu.in. system

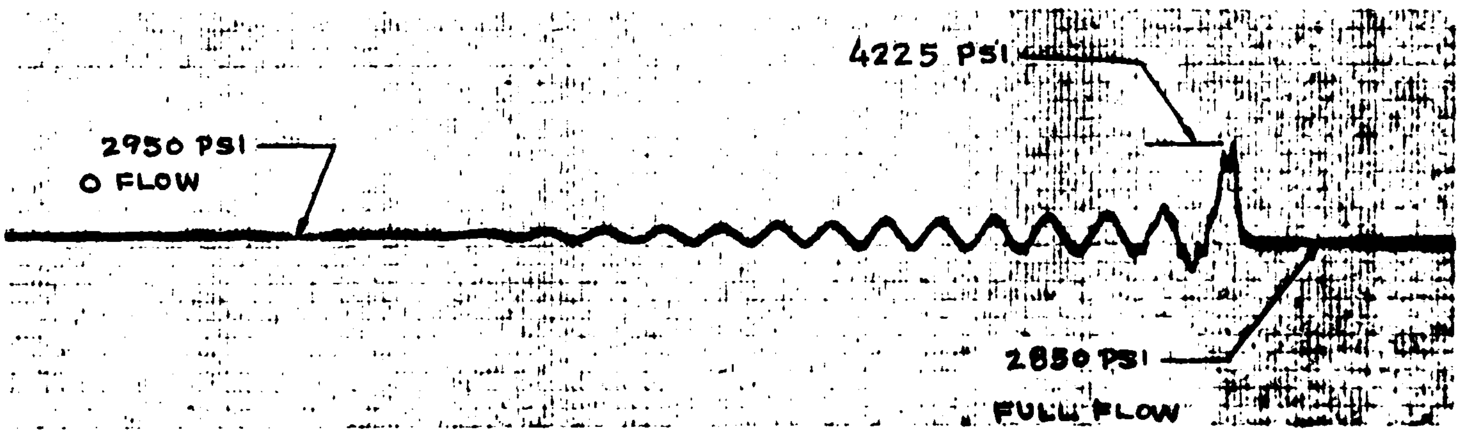


Figure 57b. Pressure transient due to a sudden valve closure while pumping AO-8 fluid into a 50-cu.in. system

Figure 57. Fifty-hour pump transient pressures at 7000 rpm

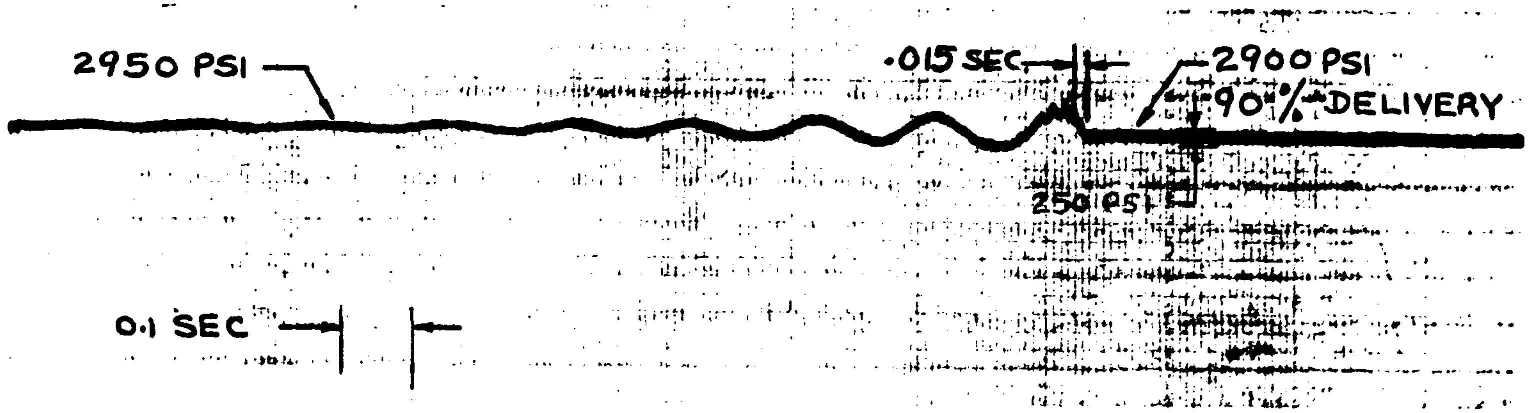


Figure 58a. Pump response following a sudden valve closure while pumping AO-8 fluid into a 250-cu.in. system

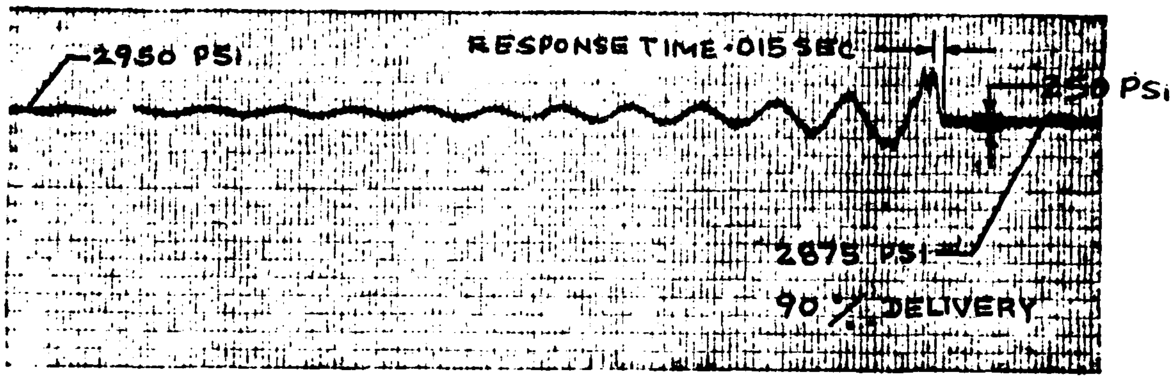


Figure 58b. Pump response following a sudden valve closure while pumping AO-8 fluid into a 50-cu.in. system

Figure 58. Fifty-hour pump response at 3500 rpm following a reduction in flow demand

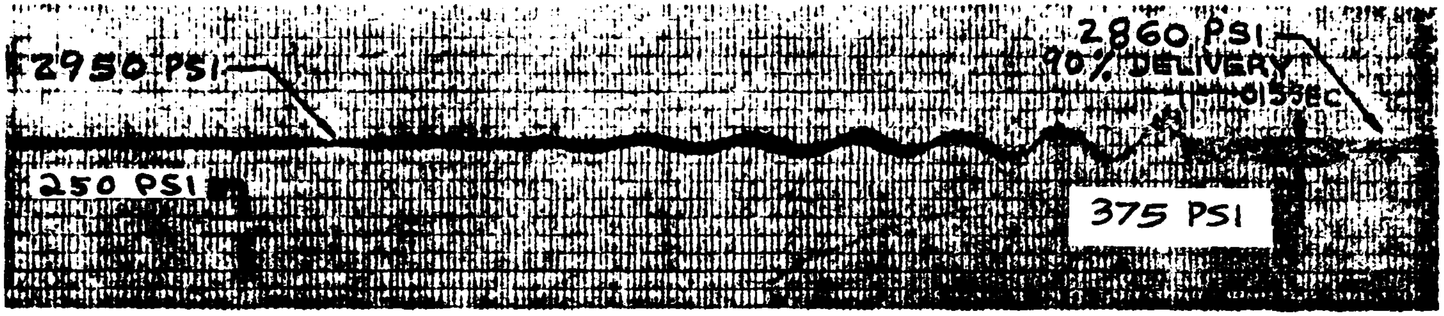


Figure 59a. Pump response following a sudden valve closure while pumping AO-8 fluid into a 250-cu.in. system

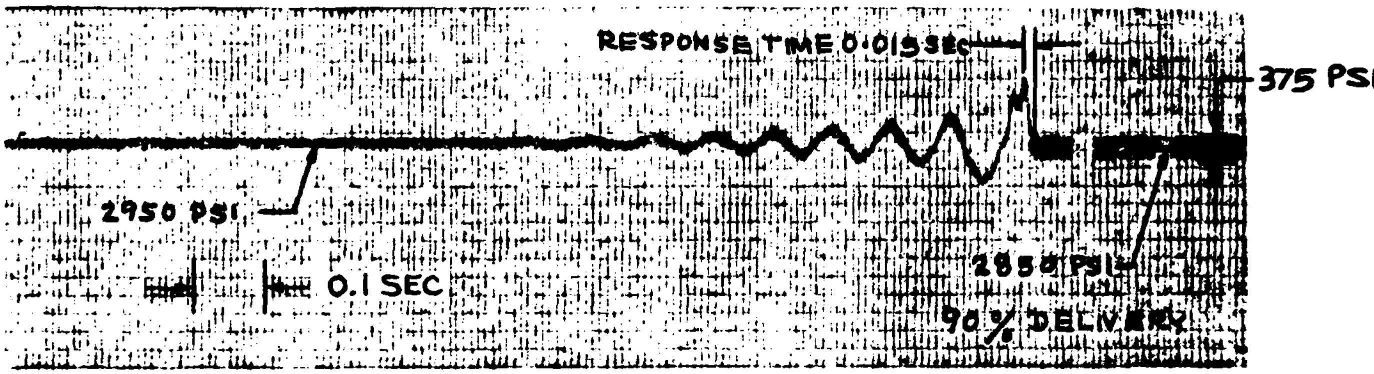


Figure 59b. Pump response following a sudden valve closure while pumping AO-8 fluid into a 50-cu.in. system

Figure 59. Fifty-hour pump response at 5250 rpm following a reduction in flow demand

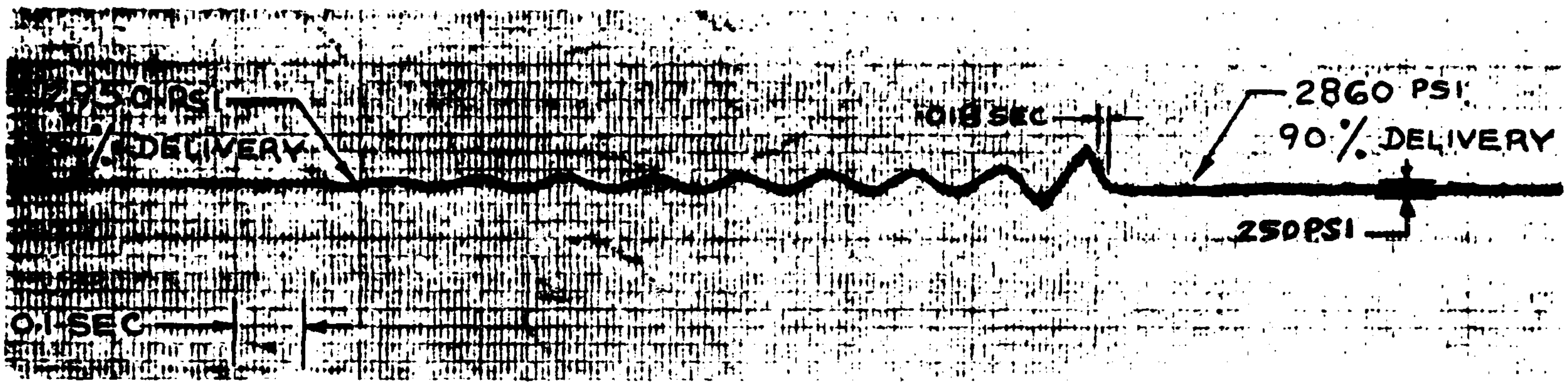


Figure 60a. Pump response following a sudden valve closure while pumping AO-8 fluid into a 200-cu.in. system

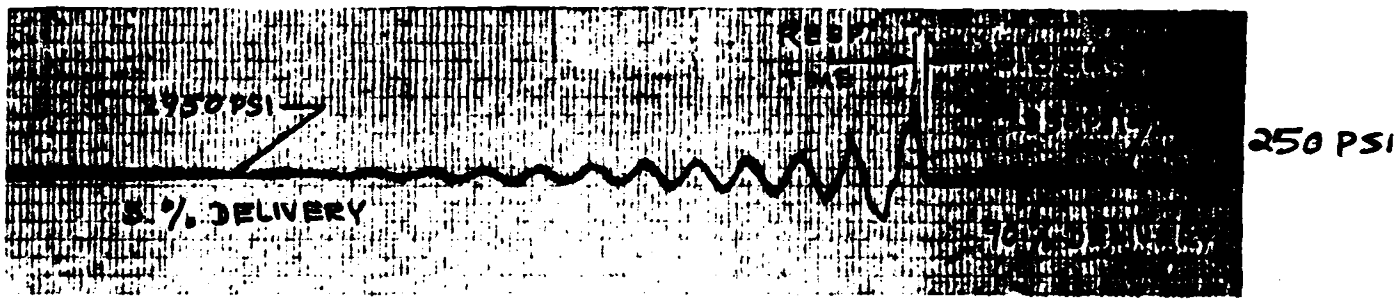


Figure 60b. Pump response following a sudden valve closure while pumping AO-8 fluid into a 50-cu.in. system

Figure 60. Fifty-hour pump response at 7000 rpm following a reduction in flow demand

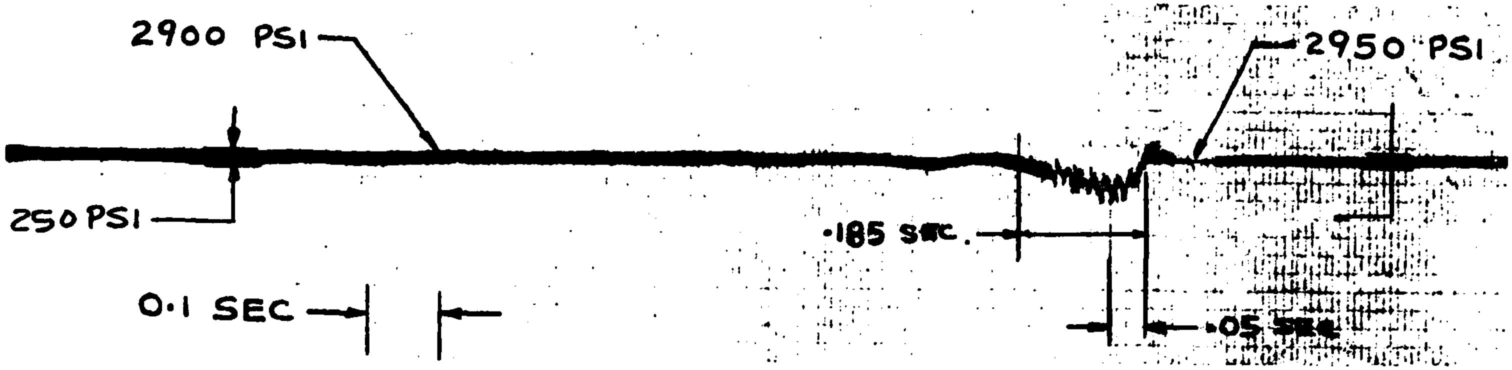


Figure 61a. Pump response following a sudden valve opening while pumping AO-8 fluid into a 250-cu.in. system

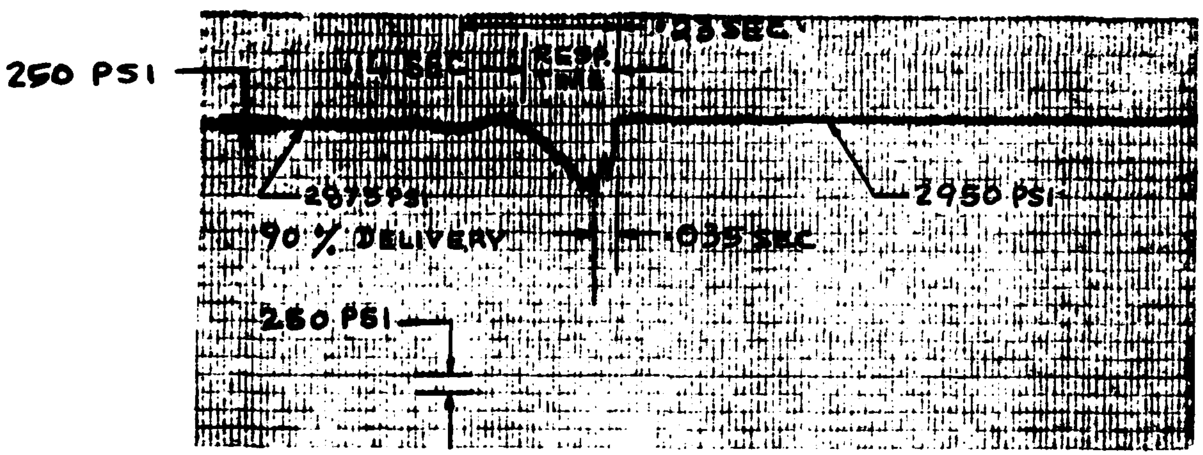


Figure 61b. Pump response following a sudden valve opening while pumping AO-8 fluid into a 50-cu.in. system

Figure 61. Fifty-hour pump response at 3500 rpm following an increase in flow demand

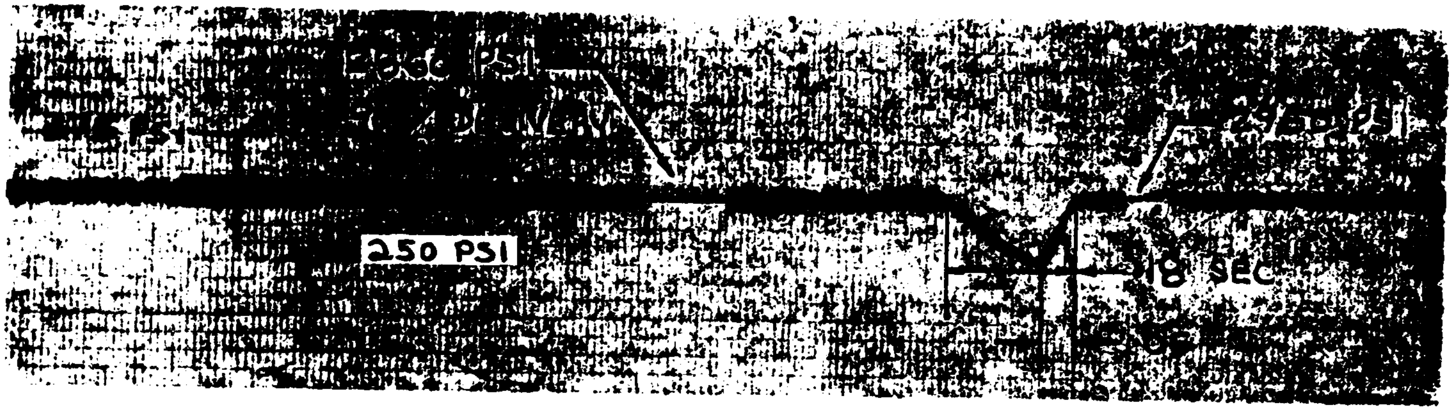


Figure 62a. Pump response following a sudden valve opening while pumping AO-8 fluid into a 250-cu.in. system

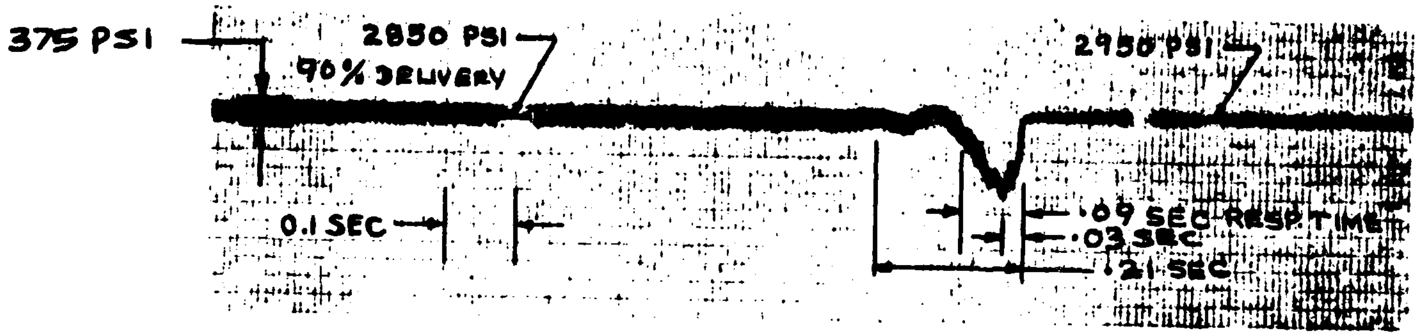


Figure 62b. Pump response following a sudden valve opening while pumping AO-8 fluid into a 50-cu.in. system

Figure 62. Fifty-hour pump response at 5250 rpm following an increase in flow demand

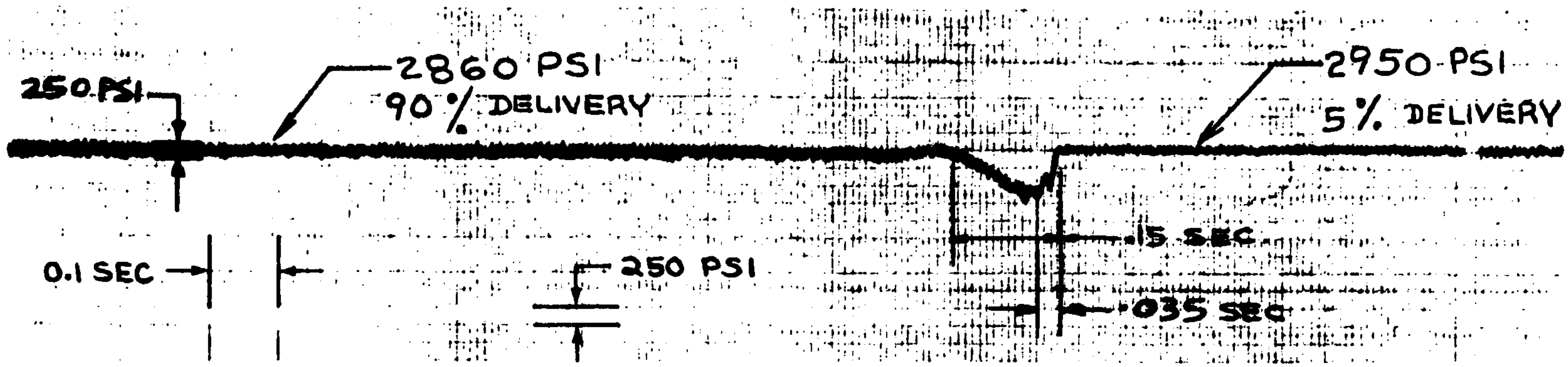


Figure 63a. Pump response following a sudden valve opening while pumping AO-8 fluid into a 250-cu.in. system

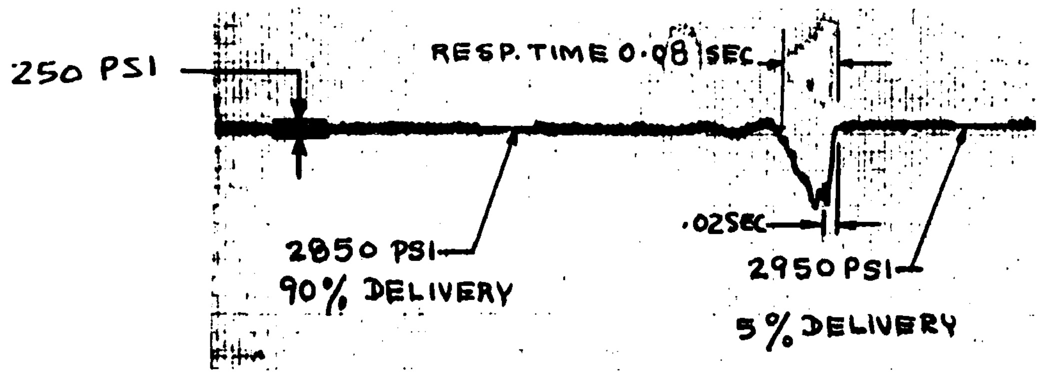


Figure 63b. Pump response following a sudden valve opening while pumping AO-8 fluid into a 50-cu.in. system

Figure 63. Fifty-hour pump response at 7000 rpm following an increase in flow demand

TABLE 23 RESPONSE AND PRESSURE PULSATION SUMMARY WITH 250 CUBIC INCH COMPLIANCE VOLUME

DATE	PUMP RPM	PRESSURE PSIG IN	PRESSURE PSIG OUT	PRESSURE PSIG CASE	TEMPERATURE °F IN	TEMPERATURE °F OUT	TEMPERATURE °F CASE	FLOW GPM	RESPONSE TIME SEC. 90-5%	RESPONSE TIME SEC. 5-90%	MAXIMUM PRESSURE PULSATION AMPLITUDE (PSI) 90-5%	MAXIMUM PRESSURE PULSATION AMPLITUDE (PSI) 5-90%
4-6-79	3500	78	2850	93	180	187	206	10.62		(1)		
4-6-79	3500	78	2900	91	192	198	220	9.47	.015	.185/.05	250	250
4-6-79	3500	79	2950	86	175	181	203	1.23				
4-6-79	5250	76	2850	100	203	211	239	16.0				
4-6-79	5250	76	2860	99	198	204	229	14.4	.015	.18/.05	375	375
4-6-79	5250	79	2950	81	183	192	221	1.28				
4-7-79	7000	74	2850	111	229	236	270	20.8				
4-7-79	7000	75	2860	107	221	228	262	18.8	.018	.15/.035	250	250
4-7-79	7000	80	2950	89	210	219	252	1.4				

(1) Response time at left in column is as per MIL-P-19692B definition

Response time at right in column is as per MIL-P-19692C definition

TABLE 24 RESPONSE AND PRESSURE PULSATION SUMMARY WITH 50 CUBIC INCH COMPLIANCE VOLUME

DATE	PUMP RPM	PRESSURE PSIG IN	PRESSURE PSIG OUT	PRESSURE PSIG CASE	TEMPERATURE °F IN	TEMPERATURE °F OUT	TEMPERATURE °F CASE	FLOW GPM	RESPONSE TIME SEC. 90-5%	RESPONSE TIME SEC. 5-90%	MAXIMUM PRESSURE PULSATION AMPLITUDE (PSI) 90-5%	MAXIMUM PRESSURE PULSATION AMPLITUDE (PSI) 5-90%
4-11-79	3500	88	2850	104	215	221	231	10.62		(1)		
4-11-79	3500	89	2875	101	217	222	232	9.09	.015	.14/.035	250	250
4-11-79	3500	89	2950	95	207	215	225	0.5				
4-11-79	5250	88	2850	110	240	247	265	15.37				
4-11-79	5250	87	2850	105	220	228	248	13.94	.015	.09/.03	375	375
4-11-79	5250	87	2950	94	184	193	218	0.75				
4-11-79	7000	86	2850	124	240	247	279	20.40				
4-11-79	7000	88	2850	120	227	234	267	18.36	.018	.08/.02	250	250
4-11-79	7000	88	2950	110	188	198	236	1.00				

(1) Response time at left in column is as per MIL-P-19692B definition

Response time at right in column is as per MIL-P-19692C definition

3.2.1.4.3.2.1 Response Time During Sudden Valve Closures

As shown below, the response times with AO-8 fluid during sudden valve closures were identical for each respective speed condition with and without the added 200 cubic inch compliance volume; and, they were almost identical to those obtained with the 5606 pump.

Pump Speed (rpm)	Response Time with AO-8 Fluid		Response Time
	$250\ in^3$ vol. (sec.)	$50\ in^3$ vol. (sec.)	5606 Fluid (sec.)
3,500	.015	.015	.0156
5,250	.015	.015	.0136
7,000	.018	.018	.020

It should be noted that MIL-P-19692B specifies that the response time (defined as the time from when the pressure starts to rise above full-flow pressure to the time when peak pressure is reached) shall not exceed .05 seconds. MIL-P-19692C specifies that that time period, designated T1, shall not exceed the value specified in the specification for the particular pump model. Figure 1 therein shows that the time from start of pressure rise to the time when the rated discharge pressure is first crossed is typically .05 seconds. In all cases, both requirements were met.

3.2.1.4.3.2.2 Response Time During Sudden Valve Openings

As shown below, the response times with AO-8 fluid during sudden valve openings were longest when discharging into the 250 cubic inch volume system; and, the response times when discharging into the 50 cubic inch system were longer than for the 5606 pump discharging into the 8-10 cubic inch system.

Pump Speed (rpm)	Response Time with AO-8 Fluid		With 5606 Fluid
	$250\ in^3$ vol. (sec/sec)	$50\ in^3$ vol. (sec/sec)	$10\ in^3$ vol. (sec/sec)
3,500	.185/.05	.14/.035	.046/.017
5,250	.18/.05	.09/.03	.035/.014
7,000	.15/.035	.08/.02	.030/.013

In each case, the response time per the MIL-P-19692B definition is shown to the left of the hash mark, and the response time per the MIL-P-19692C definition is shown to the right. MIL-P-19692B specifies that the response time (defined as the time from when the pressure starts to fall below the rated discharge pressure until it returns from its minimum value to the mean maximum full-flow pressure) shall not exceed .05 seconds. The 5606 pump met this requirement but the AO-8 pump did not.

MIL-P-19692C specifies that the response time T2 (shown as the time from when the pressure starts to fall below the rated discharge pressure until it reaches its minimum value) shall not exceed the value in the model specification. Figure 2 therein shows that the time from when the pressure starts to fall below the rated discharge pressure until it returns from its minimum value to the mean maximum full-flow pressure is typically .05 seconds; but, contrary to MIL-P-19692B, that is a not a requirement.

From the measurements taken, it appears that the response time with AO-8 fluid would be somewhat longer than the response time with MIL-H-5606 fluid in a system with equal compliance volume.

3.2.1.4.3.3 Pressure Pulsations

The peak-to-peak pressure pulsation amplitudes measured during the response time tests are tabulated in Tables 23 and 24. The peak amplitudes measured during the qualification test with MIL-H-5606 are tabulated in Figure D-9 in Appendix D, and the results from both tests are summarized below. With the AO-8 fluid, the maximum values were all obtained at the maximum flow condition (90% of full flow) and were identical with the 250 cubic inch and the 50 cubic inch compliance volumes. The flow conditions where the maximum peaks were obtained with MIL-H-5606 fluid are also noted. In all cases, the peak amplitudes were well within the specified limit of plus and minus 10% (600 psi).

Pump Speed	Max. Peak-to-Peak Pulsation Amplitude			
(rpm)	With AO-8 Fluid		With 5606 Fluid	
	Peak Pulse (psi)	Flow Rate (% of full)	Peak Pulse (psi)	Flow Rate (% of full)
3,500	250	90	275	50
5,250	375	90	320	zero
7,000	250	90	475	100

3.2.1.4.4 Endurance Tests

Both the normal-load and the overload endurance test runs were completed without incident. During the post-test inspection following the normal-load test, there was some concern about the increase in looseness of piston-shoe ball joint which led to the conservative stepped increase to the 25% overspeed condition as noted in Section 3.2.1.3.4.2 and Table 22. However, no evidence of caviation by noise signature or discharge pressure pulsation was detected.

3.2.1.5 Pump Inspections

Three teardown inspections were made wherein the pump was disassembled and visually and dimensionally examined, once during installation of the PNF pump seals, and twice during the 50-hour test, to determine if any significant wear or deterioration had occurred during the test. The findings were as follows:

3.2.1.5.1 Pre-Test Seal Installation Inspection

The "new" pump was disassembled for removal of MIL-H-5606 break-in fluid from pump parts and for installation of the PNF pump seals. The pump at this time had accumulated approximately 8 hours of break-in operation at Sperry-Vickers. Inspection of the pump components revealed the following:

a. Black deposits in the suction and discharge cavities of the valve block.

b. Slight darkening of bronze surfaces not subject to wear, such as the barrel outside diameter, piston shoes, and nutating plate.

c. Very light wear on the valve block surface.

d. Several pistons had "heat" spots on the piston diameters, considered normal by Sperry-Vickers.

3.2.1.5.2 Inspection Following Normal-Load-Endurance Testing

At this point, a total of over 40 operating hours had been accumulated. The visual and dimensional examination revealed very little degradation. No significant marking of the wear surfaces was found, and the following evidence was noted:

a. Hard black deposit in the suction and discharge cavities of the valve block.

b. Reddish coloration on the top of the bronze barrel.

c. No evidence of cavitation on the valve block valve surface nor the barrel piston ports.

d. Brown .05 diameter spots in two of the barrel piston bores.

e. Outer surface of bronze barrel had a mottled brown appearance.

f. Pistons and most bores had "heat" spots. Not unusual in high temprature applications according to the Sperry-Vickers' personnel.

g. Piston to shoe swaged ball joints had loosened between .001 to .002 since the test started, but all were still within the new manufacturing tolerance.

h. The bronze shoe lube dams had no evidence of wear but did have a mottled coloration. One shoe had a small chunk (salt grain size) missing from the inner wear pad.

i. Two of the shoe lube supply holes had slight evidence of erosion.

j. The hanger wear plate had a polishing in the discharge and suction shoe sliding areas. The polishing had a wavy pattern suggesting shoe chattering but no such evidence was visible on the shoes.

k. The hold-down-plate retainer ring had a shiny area corresponding to the discharge pressure region with dark black streaks on either side.

l. The compensator piston had several small (.05 diameter) sized heat or wear spots toward each end of sliding diameter.

m. The external shaft and shaft seal area had a significant deposit of a black tar-like substance that originated as fluid escaping through the carbon shaft seal. Samples were taken of the substance for analysis by the Materials Laboratory. The seal was not removed at this teardown.

n. The bearings were checked for roughness and none was found.

o. All the elastomeric seals (PNF) were very soft.

p. The housing seal had grown in length and could not be immediately installed in its groove. However, after it was air dried for a few days, it did return to its original size.

The pump was reassembled using the original seals except for the square-cross-sectioned upper-to-lower housing seal which had grown in length such that it wouldn't fit in the groove. Although there were several unexplained findings, they were not considered harmful to the pump operation; and, it was decided to continue the test through the overload-endurance cycling portion until a total of 50 operating hours had been accumulated.

The pump shaft was turned by hand and found to have a very high breakaway torque. The pump was partially disassembled, inspected and reassembled with a similar result. After it was reassembled again, and turned several times, the breakaway torque had reduced to an acceptable level. Reasons for the initial high torque are unknown.

3.2.1.5.3 Inspections Following Overload-Endurance Testing

At this point, a total of approximately 50 operating hours, including nearly seven hours at overpressure and overspeed conditions, had been accumulated. The initial inspection at Boeing Wichita revealed very few differences from the previous inspection. Following the on-site inspection, the pump was shipped to the Sperry-Vickers plant at Jackson, Mississippi for a more complete disassembly and metallurgical inspection. In addition, materials tests were conducted by the Materials Laboratory.

3.2.1.5.3.1 Findings of the On-Site Inspection at Boeing Wichita

Very few differences from the previous inspection were found and the following notable evidence was detected.

a. Shoe/piston ball joint looseness was unchanged.

b. Splines showed no significant wear.

c. Slight fluid leakage along with more of the tar like substance was emitted from the shaft seal. Under magnification, the carbon sealing land had a buildup of ridges or grooves in the outer third of the sealing land.

d. Shaft seal grommet had no visible degradation.

e. Housing square-sectioned seal left several small chunks in the housing groove when removed.

f. Port-cap suction and discharge passages had a reduction in the amount of black covered area and there were visible tiny cavities and white specks in the casting where the black deposite had been cleaned away.

g. The system case-drain filter element looked like new and the bowl fluid contained a small amount of black and bronze particles.

h. Several piston shoes had small chunks (salt-grain size) missing from the inner land edges.

i. The system fluid had a light yellowish-tan coloration.

3.2.1.5.3.2 Findings of the Inspection at Sperry-Vickers

Sperry-Vickers engineers reported that a visual examination, without complete disassembly of such items as the ball thrust bearing from the drive shaft, revealed a thin deposit of a dark substance on the bronze parts of the pump, sluggish rotation of the drive shaft, hot spots on the pistons, erosion of the piston shoes, and a black gummy deposit in the housing adjacent to the shaft seal. In a more detailed examination, which included a more complete disassembly and examination of the bearings by the MPB Corporation, the following additional problems were reported:

a. The drive shaft in the area covered by the bearing inner race was deeply etched.

b. The drive shaft ball bearing showed evidence of inadequate lubrication.

c. The yoke pintle bearings also showed evidence of inadequate lubrication.

d. The yoke pintles showed evidence of rust.

The complete Sperry-Vickers inspection report, including their metallurgical evaluation, is documented in Appendix E. In regard to the reported oxidation corrosion, it should be noted that the steel parts are primarily non-corrosion-resistant alloys; and, that the parts probably rusted after having been dewetted of all protective fluid films, and were attacked by the moist ambient air prevalent in Jackson, Mississippi.

3.2.1.5.3.3 Findings of Materials Tests Conducted by the Materials Laboratory

The Materials Laboratory Nonmetallic Materials Division's Fluids, Lubricants, and Elastomers Branch had seven pump parts analyzed by the Auger Spectroscopy Group at the University of Dayton. The results are reported in Appendix F.

The Materials Laboratory analyzed samples of used hydraulic fluid taken during the test and two black tarry samples taken from the coupling shaft spline. In addition, they ran an errosion oxidation test using copper metal test specimens in order to better understand the reason for the reddish deposit on the bronze parts of the pump. These results are reported in Appendix C.

3.2.1.6 Fifty-Hour Pump Test Conclusions

The following conclusions can be drawn from the data measured during this fifty-hour test.

a. No redesign of the pump is required to obtain full rated delivery flow, with acceptable power input levels, with the AO-8 fluid.

b. Higher inlet pressures are required with AO-8 fluid than with MIL-H-5606 fluid to avoid cavitation and the resultant reduction in flow delivery and possible damage. This is due to its higher density and absolute viscosity; and, some system weight penalties will probably be incurred to provide higher reservoir pressures, larger suction lines, or suction line boost pumps. Larger pump inlet porting and fluid inlet passages may effectively reduce inlet flow losses and the need for higher inlet pressures.

c. Transient discharge pressure peaks will probably be accpetable (within the 135% limit) for all but very close-coupled systems which may require some additional fluid compliance volume.

d. The pump response to sudden valve closures (changing from high-flow to low-flow delivery) appears acceptable. The response to sudden valve openings (changing from low-flow to high-flow delivery) was slower than with MIL-H-5606 fluid, but would be nearly comparable in systems with equal compliance volume.

e. Pressure pulsations are comparable to those obtained with MIL-H-5606 fluid and well within specified limits.

The following conclusions can be drawn from the examinations of the pump during the post-test inspections.

f. The carbon shaft seal member appeared to be adversely attacked by the AO-8 fluid.

g. The AO-8 fluid has marginal lubricity for highly loaded pump bearings.

h. The PNF elastomer seals used appear to soften too much to guarantee sealing throughout the life specified for military aircraft pumps. Perhaps they should be molded with a harder (higher durometer) compound of the PNF material.

In total retrospect, however, it was concluded that, with a few relatively minor revisions to the pump, to the fluid, and to the operating procedures, there is a reasonably good probability that the present basic pump design could successfully meet the Long Term Pump Test requirements.

3.2.1.7 Fifty-Hour Pump Test Recommendations

The following recommendations were developed by consultation with the following personnel involved with this program:

Messrs. K. E. Binns and W. B. Campbell, Aero Propulsion Laboratory
Mr. C. E. Snyder, Jr., Materials Laboratory
Messrs. K. E. Becker and N. F. Pedersen, Sperry-Vickers
Mr. W. Cassanos, Halocarbon Products
Messrs. A. Parekh and E. C. Wagner, Boeing Wichita
Messrs. D. W. Huling and E. T. Raymond, Boeing Seattle

a. The second PV3-075-15 pump for this program should be assembled and inspected (at Sperry-Vickers) with new (unused) parts of the original design, except for the carbon shaft seal member, and with the PNF seals currently available.

b. The carbon shaft seal member should be replaced with a bronze piece of the same configuration.

c. A compatable chlorofluorocarbon grease provided by Halocarbon Products, through Mr. Snyder, should be used to lubricate pump parts during assembly.

d. The pump should be run in on the Boeing Wichita test stand with a formulation of Halocarbon AO-8 fluid to which the Molyvan A lubricity and anti-wear additive tested by the Materials Laboratory has been added (as recommended by laboratory personnel).

e. A teardown and inspection should be conducted at Boeing Wichita following the break-in run being extremely careful to avoid prolonged exposure to the atmosphere.

f. The pump should then be reassembled and given an abbreviated break-in run per a procedure to be supplied by Sperry-Vickers.

g. The pump should then be run for 50 hours of endurance testing per the procedure identified in the Appendix H plan.

h. A second teardown and inspection should be conducted at Boeing Wichita; and, if further evaluation is necessary, again at Sperry-Vickers. Again, extreme care should be exercised to avoid prolonged exposure to the atmosphere.

i. If the condition is satisfactory, the long-term pump test should be continued until completion or failure.

3.2.2 Long-Term Pump Test

This test was run primarily on a second pump, modified in accordance with the conclusions reached following the fifty-hour pump test, to determine its performance and endurance life under conditions specified in MIL-P-19692B. The test fluid was Halocarbon AO-8, Batch No. 62979, with the lubricity and anti-wear additive Molyvan A (molybdenum oxysulphide dithiocarbamate) added in the quantity of less than one percent by weight.

Again, as in the fifty-hour pump test, Firestone PNF Compound 200R-211658 elastomeric seals were used. The O-rings were molded by Nichols Engineering, and the special seals were molded by Precision Rubber Products Corporation.

3.2.2.1 Test Pump

The pump utilized in this test program was a Sperry-Vickers model PV3-075-15, modified to include a bronze shaft seal element and PNF elastomer seals, which was redesignated as model PV3-075-19. The serial number of the test pump was MX-337143.

The pump was assembled at Sperry-Vickers with all new components, filled with AO-8 fluid, and shipped to Boeing Wichita for break-in and test.

3.2.2.2 Test Setups

3.2.2.2.1 *Performance/Endurance Test Setup*

The test setup utilized for all performance and endurance tests is depicted schematically in Figure 64. Equipment utilized in the test setup is tabulated in Table 25.

3.2.2.2.2 Cold-Start Setup

A smaller, more abbreviated test setup which could be more easily refrigerated to -65°F was fabricated for the cold-start test. Due to failure of the long-term test pump, this test was performed with the refurbished Sperry-Vickers PV3-075-19 pump which had been used previously in the fifty-hour pump test and in the servoactuator test described in Section 3.3. Equipment utilized in the cold-start test setup is tabulated in Table 26 and is shown schematically in Figure 65.

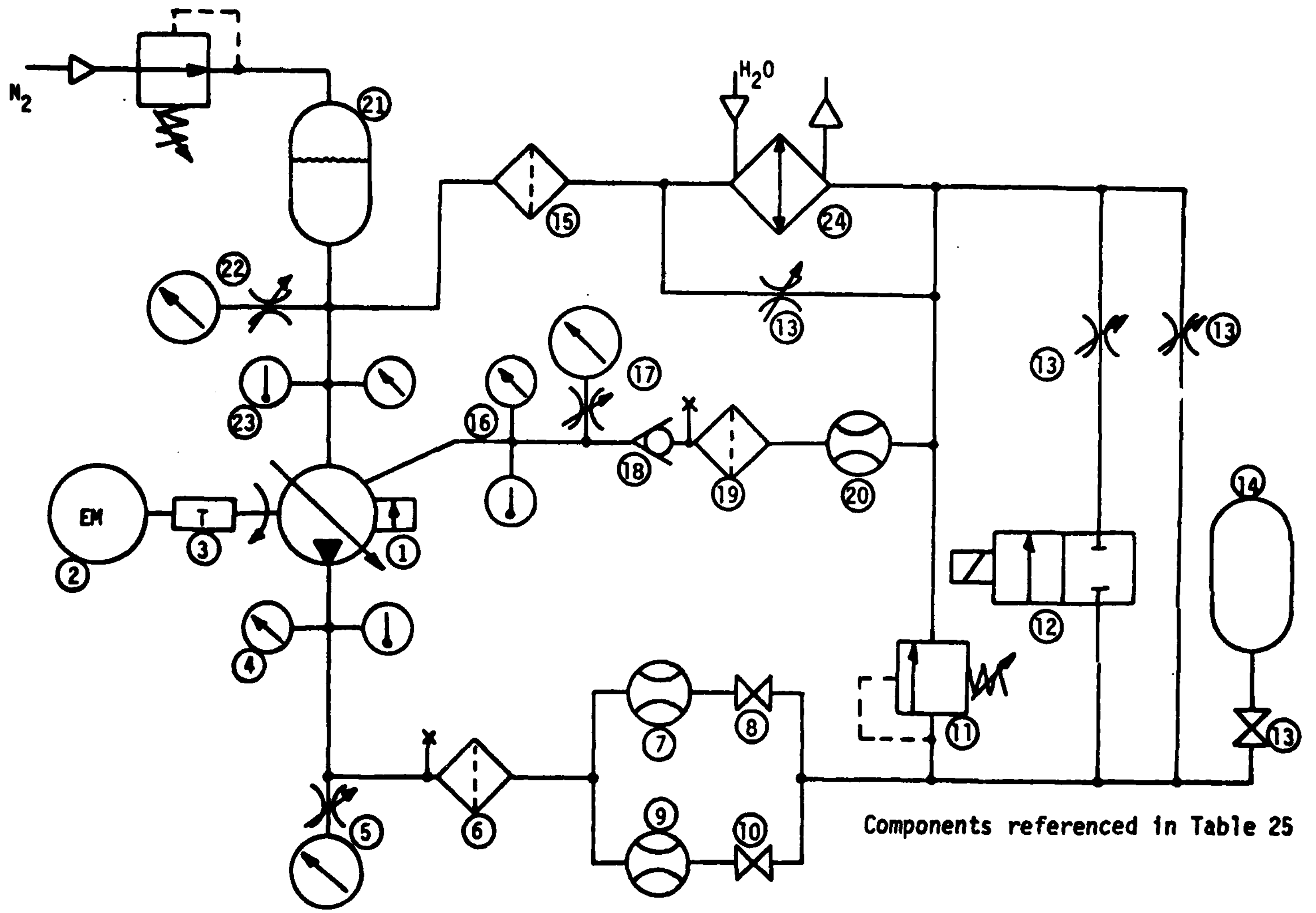
3.2.2.3 Test Performance

The long-term pump test was performed in compliance with the Long-Term Pump Test Plan, Appendix H, over the period of 15 August 1979 to 21 January 1980. Testing was interrupted between 6 September 1979 and 17 October 1979 for repair of the pump.

3.2.2.4 Test Results

3.2.2.4.1 Rotational Torque Check

A rotational torque check was performed on the newly assembled pump



Components referenced in Table 25

Figure 64. Long-term pump endurance test setup

TABLE 25 LONG-TERM PUMP ENDURANCE TEST EQUIPMENT LIST

1. Pump, Sperry-Vickers, PV3-075-19, S/N MX-337143
2. Varidrive, U.S. Motors, VEU-GSDT, 75 HP, S/N 940865
3. Torque Transducer, Lebow Associates, 1235-107, S/N 121
4. Discharge Pressure Transducer, CEC, 4-326-008, 0-5000 psig, S/N 17413
5. Discharge Pressure Gage, Glassco, 0-5000 psig 1/4%, S/N RL4086
6. Discharge Filter Assy, Aircraft Porous Media, AD-3255-16D-11, S/N 9680-2265 (housing)
7. Discharge Flowmeter, Cox, AN8-4, S/N 22013
8. Flowmeter Shutoff Valve, Parker Hannifin, MV-830-S
9. Discharge Flowmeter, Cox, AN 12, S/N 21222
10. Flowmeter Shutoff Valve, Parker Hannifin, MV-1230-S
11. System Relief Valve, Mission Valve, 151102-2120, AND-12 x AND-16, S/N 1916-1
12. Solenoid Valve, Marotta, 280203, S/N 106
13. Load/Shutoff Valve, Parker Hannifin, MV-1230-S
14. Compliance Volume, Parker Aircraft, 2660359, 200 cu. in. (accumulator piston removed)
15. Return Filter Assy, Aircraft Porous Media, AD-3258-16Y69, S/N 0110
16. Case Drain Pressure Transducer, CEC, 4-326, 300 psig, S/N 17907
17. Case Drain Pressure Gage, Ashcroft, 0-300 psi, 5 lb. sub-division, S/N RL4059
18. Check Valve, Gar, 4CV088, AN6280-8
19. Case Drain Filter Assy, Aircraft Porous Media, AC-3258-68Y15, S/N 1760-0509 (housing)
20. Case Drain Flowmeter, Cox AN8-4, S/N 11672
21. Reservoir, Boeing Co., 35-3297
22. Inlet Pressure Gage, Roylyn Lindsay, 300 psi, 2 psi divisions
23. Inlet Pressure Transducer, CEC 4-326-0001, 0-250 psia, S/N 21942
24. Heat Exchanger, American Standard, Type BCF, Mod 5-030-05-024-003

TABLE 26 PUMP COLD-START TEST EQUIPMENT LIST

1. Test Pump, Sperry-Vickers, PV3-075-19, S/N MX-319687
2. Varidrive, U.S. Motors, VEU-GSDT, 75 HP, S/N 940865
3. Discharge Pressure Gage, Ashcroft, 0-5000 psig, 50 lb subdivision, S/N RL4060
4. Discharge Filter Assy, Aircraft Porous Media, AD-3255-16D-11, S/N 9680-2265 (housing)
5. Discharge Flowmeter, Cox, AN8-4, S/N 22013
6. Relief Valve, Mission Valve, 151102-2120, AND-12 x AND-16, S/N 1916-1
7. Load Valve, Parker Hannifin, MV-1230-S
8. Reservoir, Boeing Co., 35-3297
9. Inlet Pressure Gage, Ashcroft, 0-400 psi 2 lb subdivision, S/N USAF B-2387
10. Check Valve, Gar, 4CV 088, AN 6280-8
11. Case Drain Filter Assy, Aircraft Porous Media, AC-3258-68Y15, S/N 1760-0509 (housing)

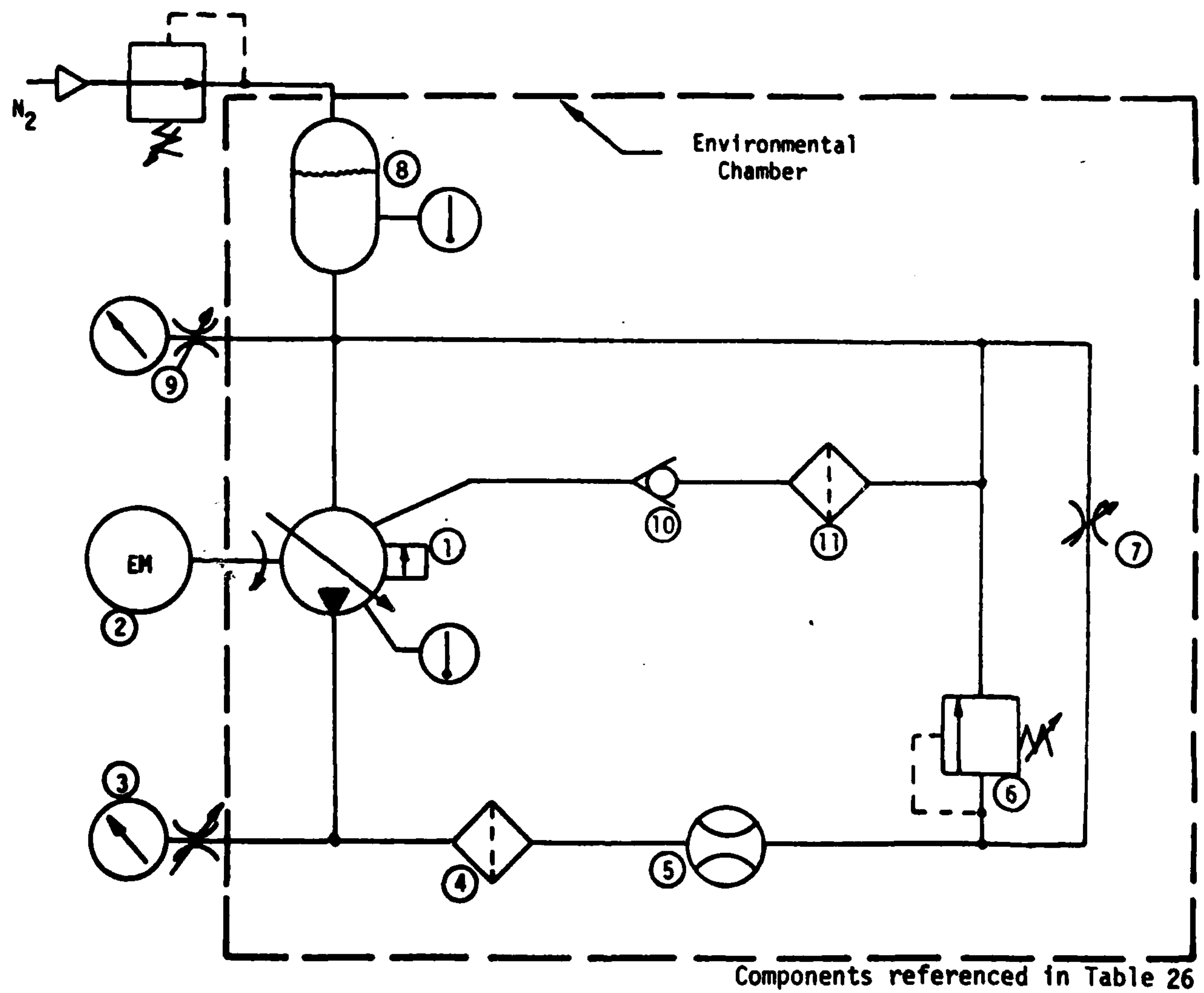


Figure 65. Pump cold-start test setup

at Sperry-Vickers on 14 August 1979, and the following values measured:

Condition	Torque (lb-in)	
	Actual	Limit
Breakaway	27.5	30
Running	25	25

3.2.2.4.2 Proof Pressure Test

A proof pressure test was performed at Wichita on 15 August. No external leakage was noted.

3.2.2.4.3 Initial Break-In Run

The break-in run was started on 16 August and discontinued on 17 August due to an unidentified pump noise and to a 60°F temperature differential between the case drain and pump inlet. Pump operating time at shutdown was 4:00 hours. Compensator instability was the suspected cause of the unusual pump noise.

3.2.2.4.4 Teardown Inspection

The pump was disassembled on 22 August by Sperry-Vickers personnel. The head of the actuator piston was found badly scored (Figure 66). The piston head was hand lapped and the pump was reassembled with new seals.

3.2.2.4.5 Initial Break-In Run

The break-in run was resumed and successfully completed on 22 August. The 60°F inlet to case drain temperature differential was due to reduced case flow (approximately 0.44 gpm at 7000 rpm, full flow).

3.2.2.4.6 50-Hour Run

The 50-hour run was accomplished without incident over the period of 22 August to 4 September.

3.2.2.4.7 Rotational Torque Test

A rotational torque test was performed after completion of the 50-hour test. Breakaway torque was not recorded. Running torque was 30 lb-in. (limit 25 lb-in.).

3.2.2.4.8 Teardown Inspection

The pump was hand carried to Sperry-Vickers for inspection on 6 September where the following conditions were observed.

Figure 66. Long-term pump actuator piston after break-in run

3.2.2.4.8.1 Valve Block

The valve block face and all interior passages in contact with the fluid were discolored. Considerable bronze was transferred from the cylinder block Kingsbury pads (see Figure 67).

Lapping of the valve block surface to remove the bronze build-up revealed a soft area under the heaviest bronze build-up. Repair attempts failed and the valve block was replaced.

3.2.2.4.8.2 Cylinder Block

The cylinder block was discolored over all surfaces. Considerable bronze was transferred from the Kingsbury pads (see Figure 68). The cylinder bores were worn by approximately 0.0001 inch.

The cylinder block face was successfully repaired by relapping.

3.2.2.4.8.3 Drive Shaft

The drive shaft exhibited an overall discoloration except for the race for the needle bearing. The thrust bearing was rough when dry and had an overall satin gray appearance as if sand blasted or etched.

3.2.2.4.8.4 Actuator Piston

No additional damage to the piston head had occurred since the initial break-in. The piston surface near the end exhibited a 1/4 inch by 3/8 inch long area of scoring or galling, but no corresponding marks were found in the housing bore.

The actuator piston was replaced during reassembly of the pump.

3.2.2.4.8.5 Piston/Shoe Subassemblies

The piston barrels were slightly discolored and the shoe outer surfaces were highly darkened. The piston barrels had apparently increased in diameter by 0.0001 due to an unidentified coating. Two shoes were showing heavy erosion around the center hole and the outer ring of most shoes were starting to erode in the line with the slots.

All piston/shoe subassemblies exhibited 0.002-inch shoe end play except for number 7 which had 0.0045 (originally 0.002). Number 7 piston/shoe subassembly was replaced at pump build-up.

3.2.2.4.8.6 Steel Parts

All internal steel parts were discolored by a "coating" which could not be removed. Discoloration ranged from a slight straw color of the shoe wear plate to an almost black color of the yoke control springs.

3.2.2.4.8.7 Bronze Parts

All bronze pump components exhibited a dark discoloration which could be partially removed by rubbing.

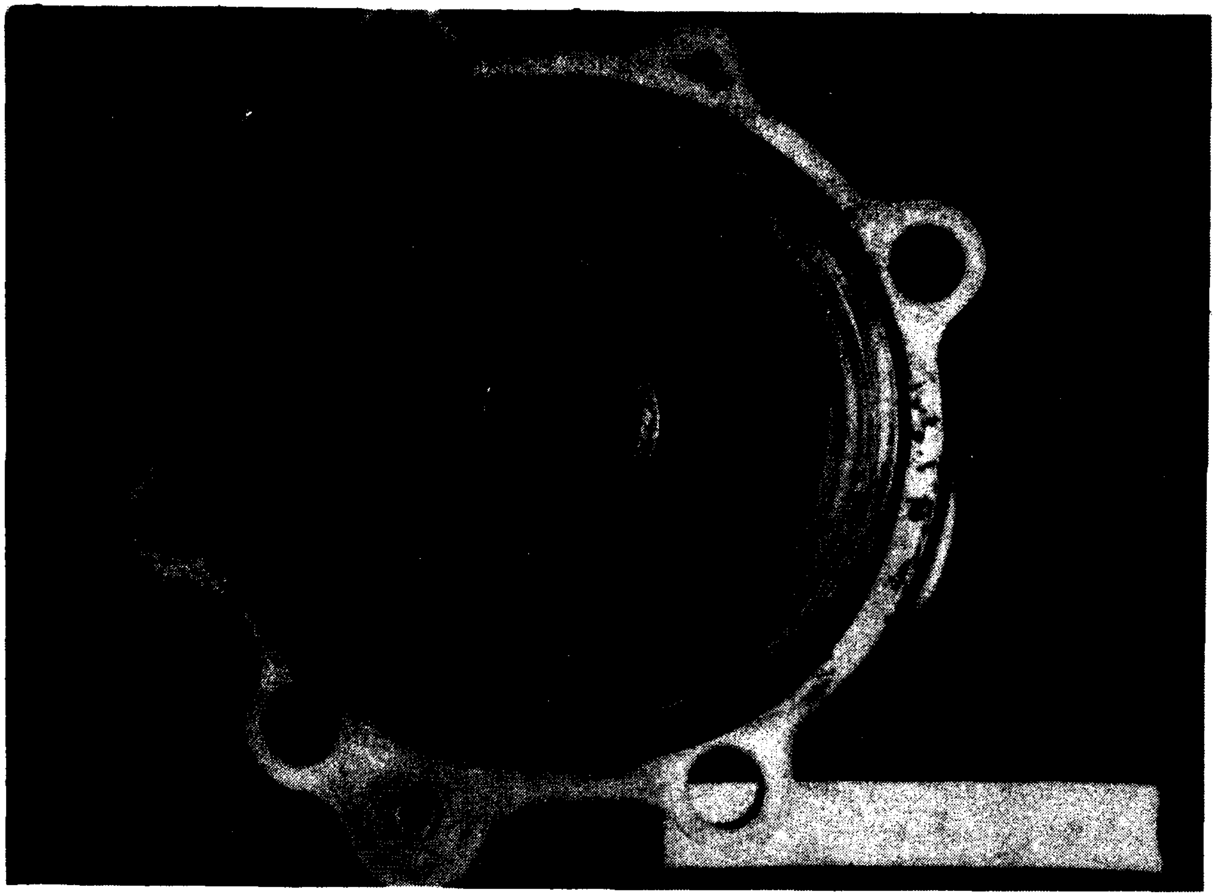


Figure 67. Long-term pump valve block after break-in run

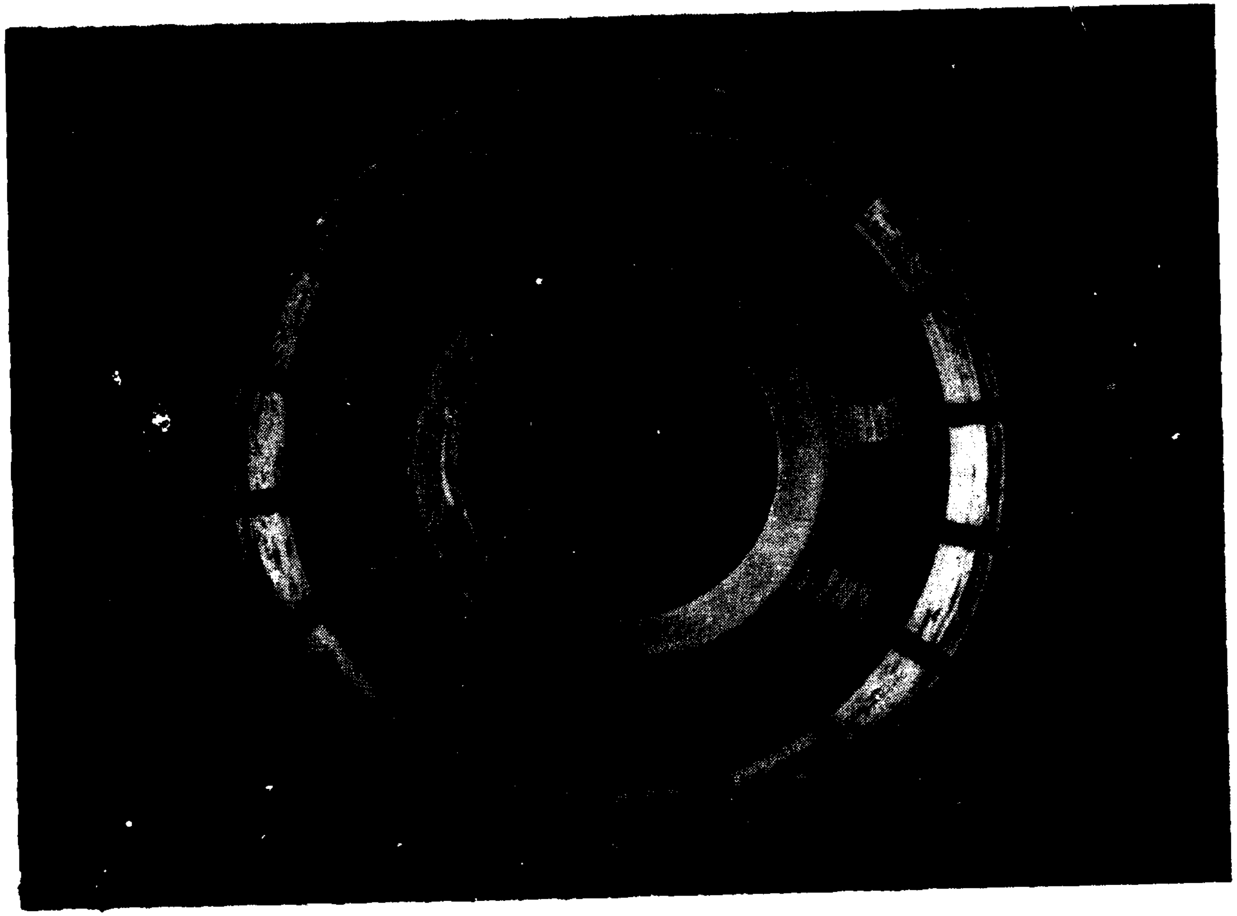


Figure 68. Long-term pump cylinder block after break-in run

3.2.2.4.8.8 Elastomers

The elastomers exhibited slight softening and were replaced at pump build-up.

3.2.2.4.9 Rotational Torque Check

The pump was returned to Wichita on 17 October after a delay in obtaining a new valve block. A rotational torque check was made and the following values measured:

Condition	Torque (lb-in)	
	Actual	Limit
Breakaway	32 first time 30 subsequently	30
Running	20	25

3.2.2.4.10 Proof Pressure Test

The proof pressure test requirement was reduced from 500 psig to 100 psig to reduce the possibility of pump case failure. No leakage was observed at 100 psig for 5 minutes or 1 foot of head for 30 minutes.

3.2.2.4.11 Abbreviated Run-In

The abbreviated run-in was accomplished on 24 October with no problems. The pump compensator was readjusted to 3025 psig with no flow at the end of the run-in.

3.2.2.4.12 Calibration Tests

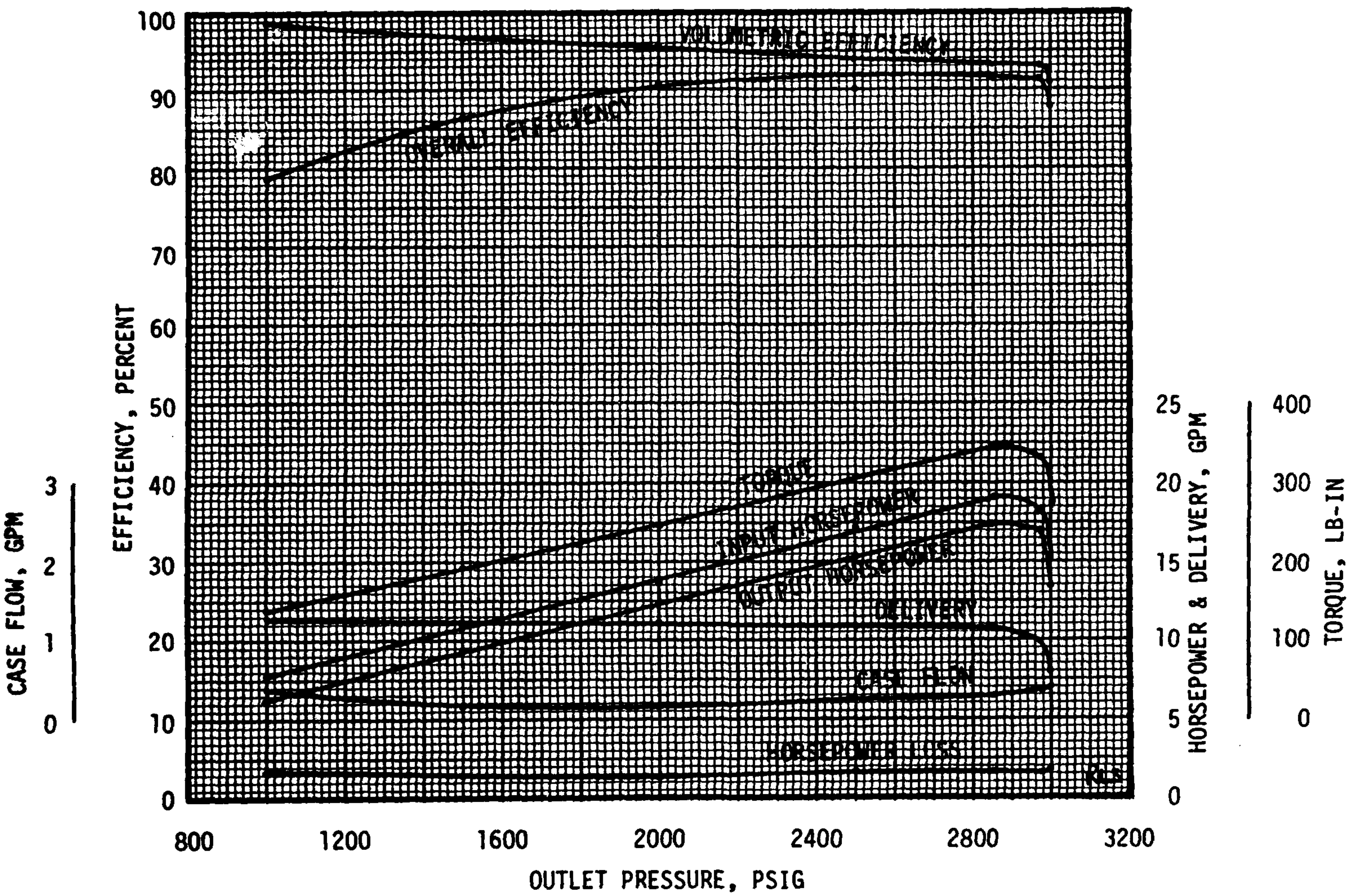
Calibration tests were performed at 3500, 5250, 7000, and 7700 rpm on 25 and 26 October.

Calibration test results are plotted on Figures 69 through 76.

3.2.2.4.13 Transient Pressure Test

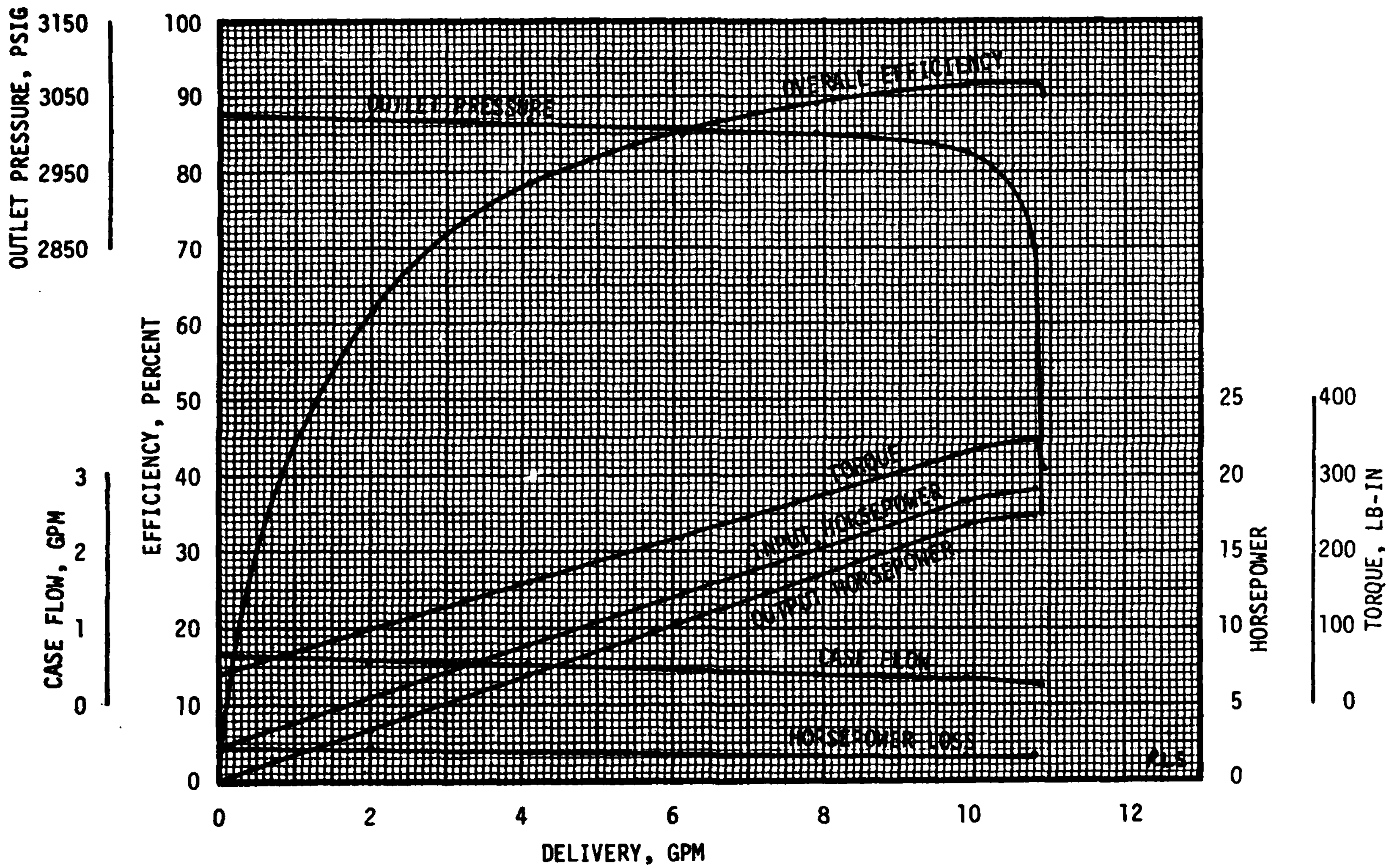
A transient pressure test was performed on 30 October at pump speeds of 3500 and 7000 rpm, both with and without the 200 cubic inch compliance volume added to the circuit.

Test results are shown in Figures 77 and 78 and are summarized in Table 27. Transient pressure at 7000 rpm was 144 percent of rated pressure without the 200 cubic inch added compliance volume, and 120 percent of rated pressure with the added compliance volume.



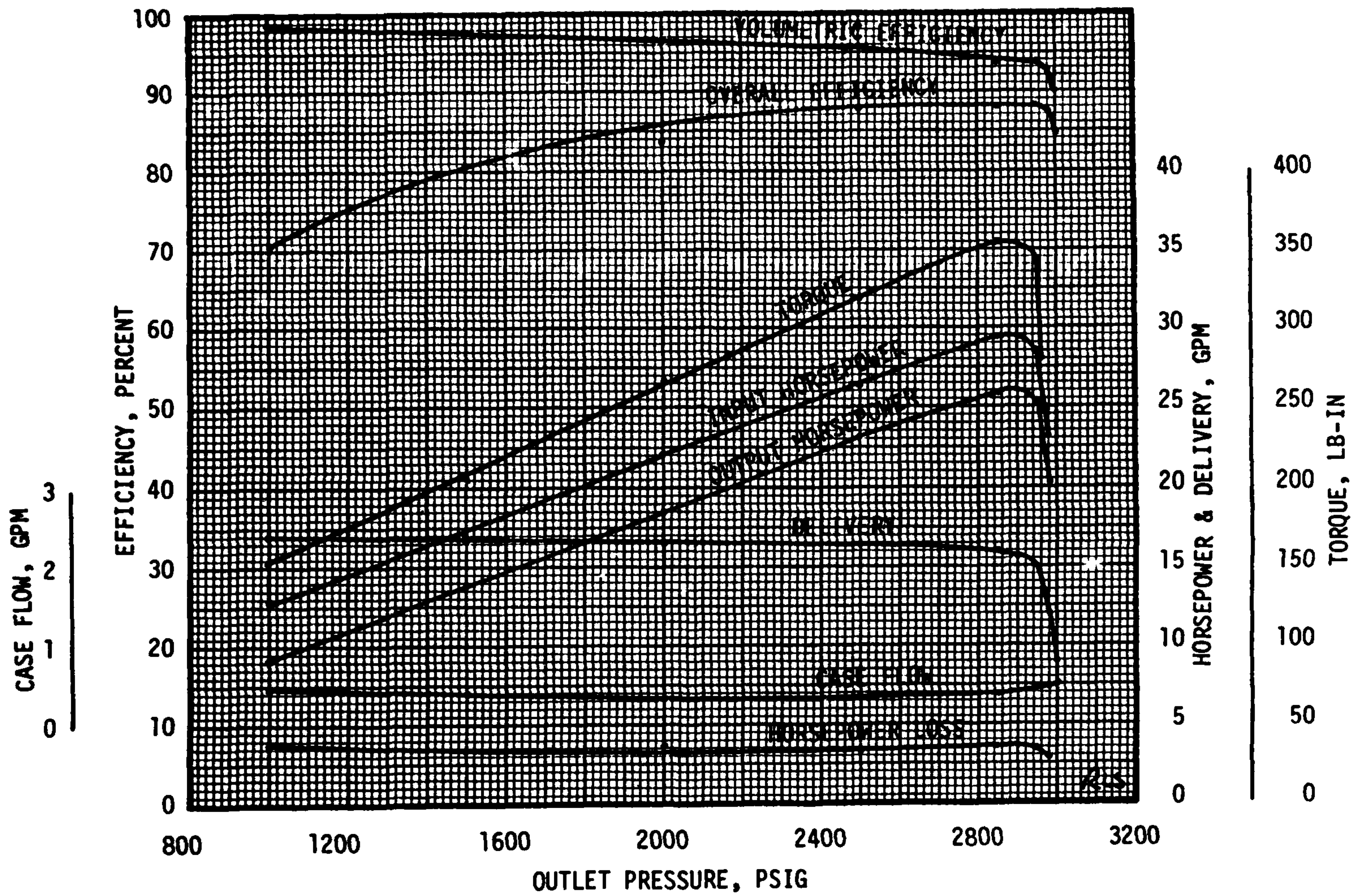
Fluid: Halocarbon AO-8   Inlet Pressure: 85 psig   Temperature: 180 ± 5F

Figure 69. Long-term pump full-flow performance at 3500 rpm



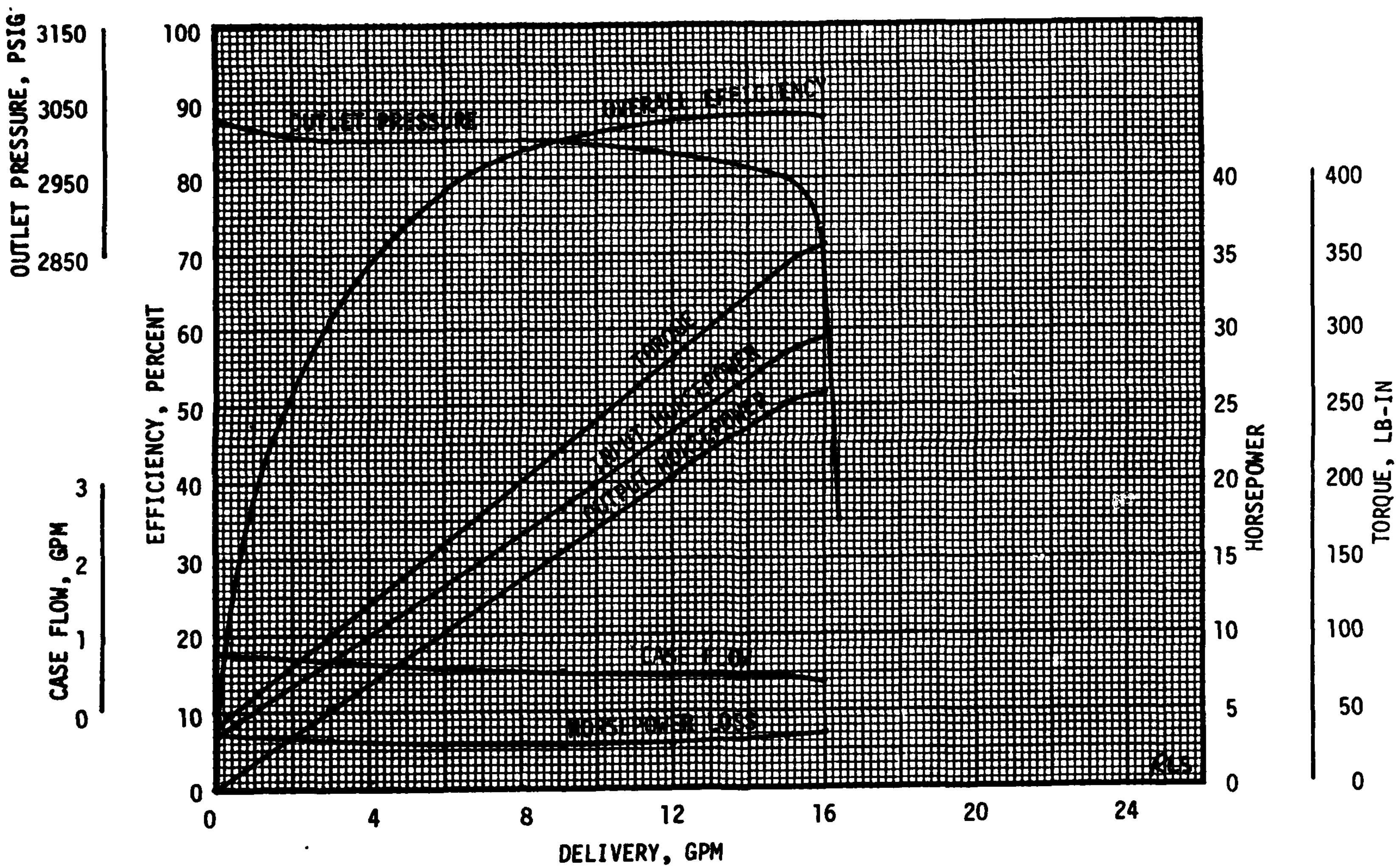
Fluid: Halocarbon AO-8 Inlet Pressure: 85 psig Temperature: 180 ± 5F

Figure 70. Long-term pump partial-flow performance at 3500 rpm



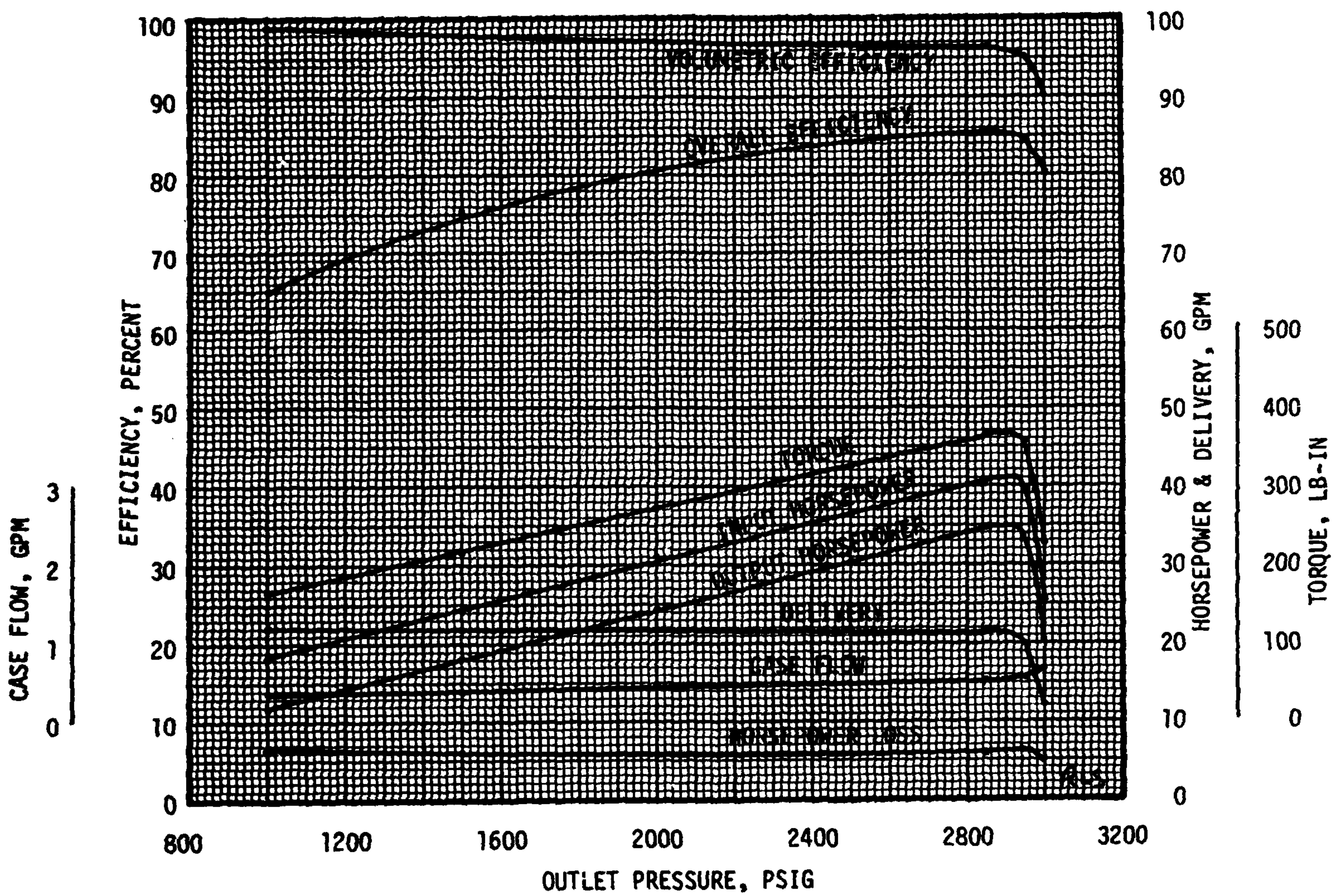
Fluid: Halocarbon AO-8 Inlet Pressure: 84 psig Temperature: 180 ± 5F

Figure 71. Long-term pump full-flow performance at 5250 rpm



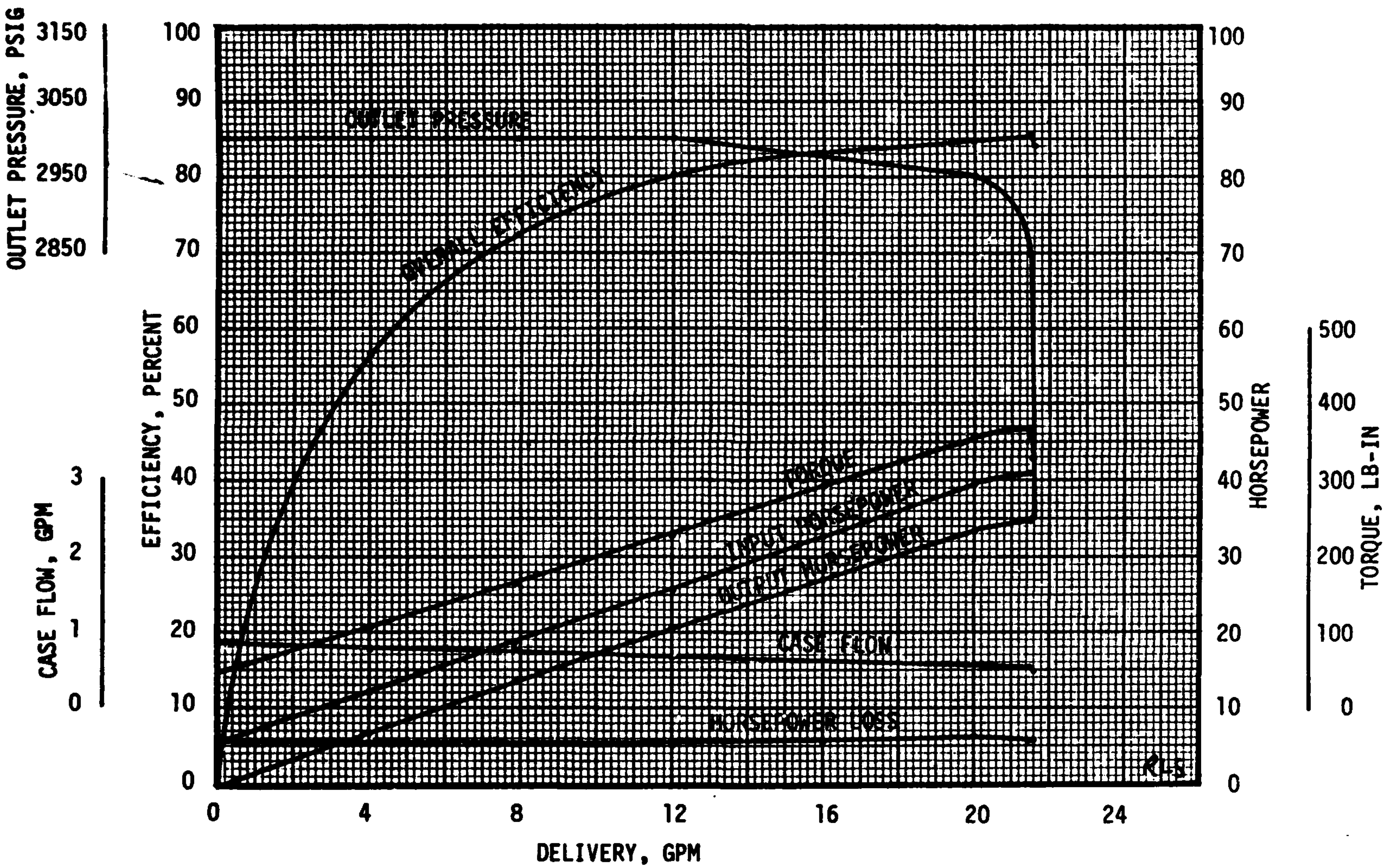
Fluid: Halocarbon AO-8 Inlet Pressure: 84 psig Temperature: 180 ± 5F

Figure 72. Long-term pump partial-flow performance at 5250 rpm



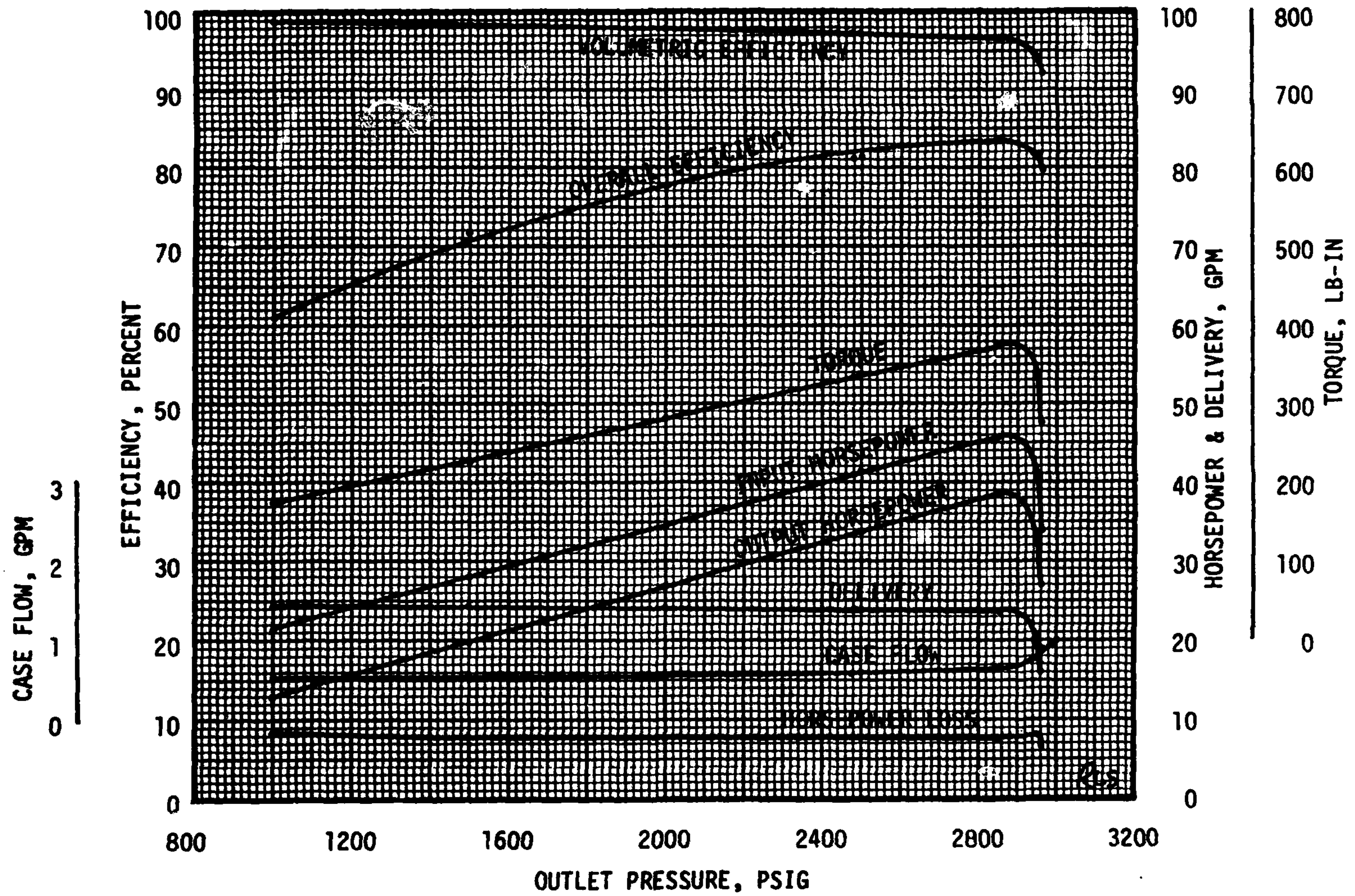
Fluid: Halocarbon AO-8 Inlet Pressure: 83 psig Temperature: 180 ± 5F

Figure 73. Long-term pump full-flow performance at 7000 rpm



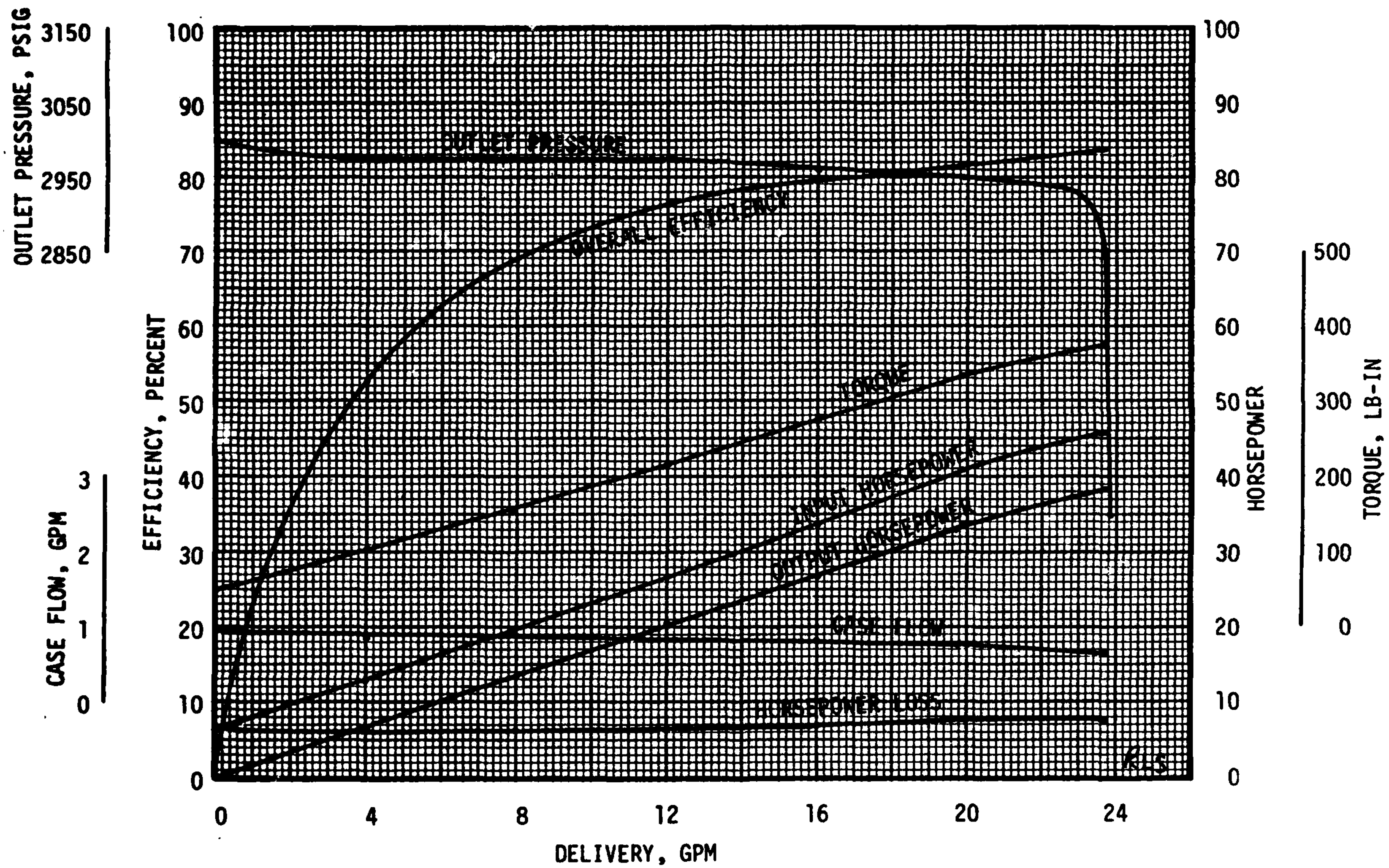
Fluid: Halocarbon AO-8 Inlet Pressure: 83 psig Temperature: 180 ± 5F

Figure 74. Long-term pump partial-flow performance at 7000 rpm



Fluid: Halocarbon AO-8 Inlet Pressure: 84 psig Temperature: 180 ± 5F

Figure 75. Long-term pump full-flow performance at 7700 rpm



Fluid: Halocarbon AO-8 Inlet Pressure: 84 psig Temperature: 180 ± 5F

Figure 76. Long-term pump partial-flow performance at 7700 rpm

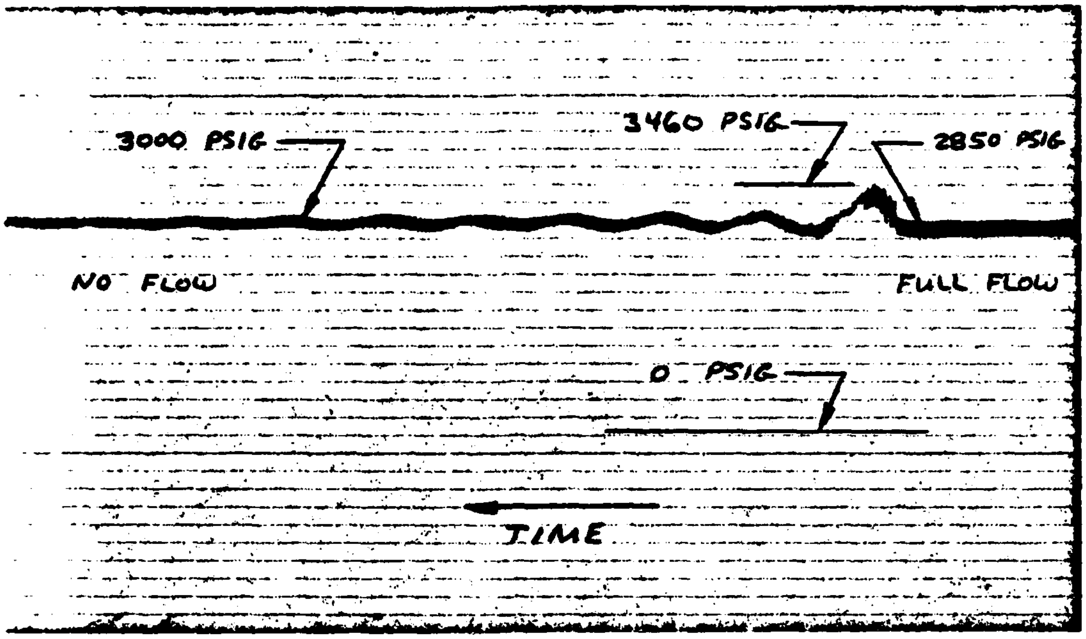


Figure 77a. Pressure transient due to a sudden valve closure while pumping AO-8 fluid into a 250-cu.in. system

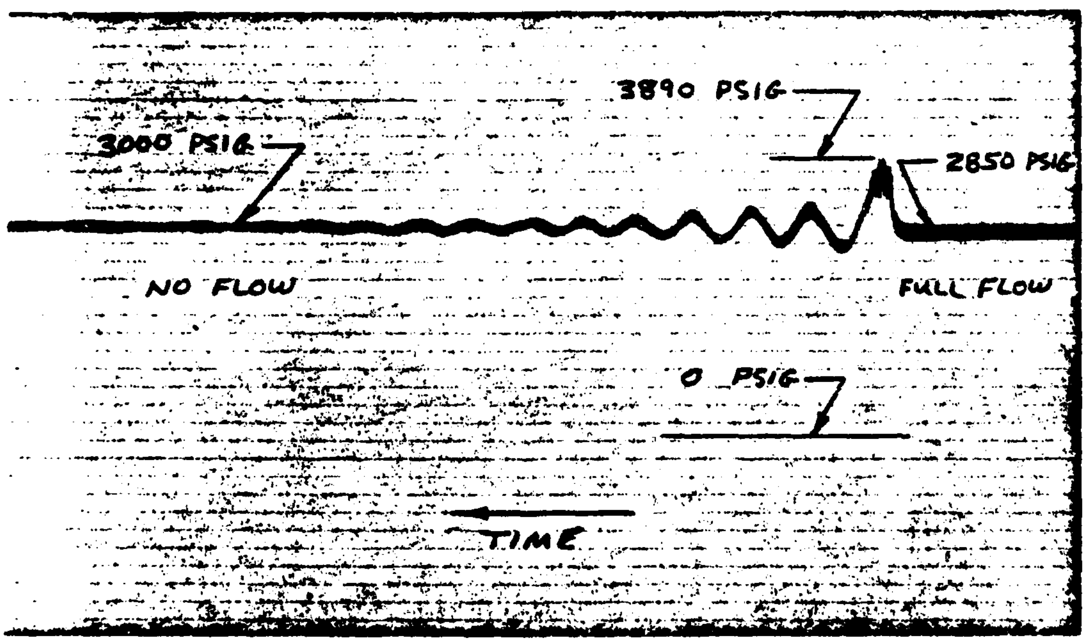


Figure 77b. Pressure transient due to a sudden valve closure while pumping AO-8 fluid into a 50-cu.in. system

Figure 77. Long-term pump transient pressures at 3500 rpm

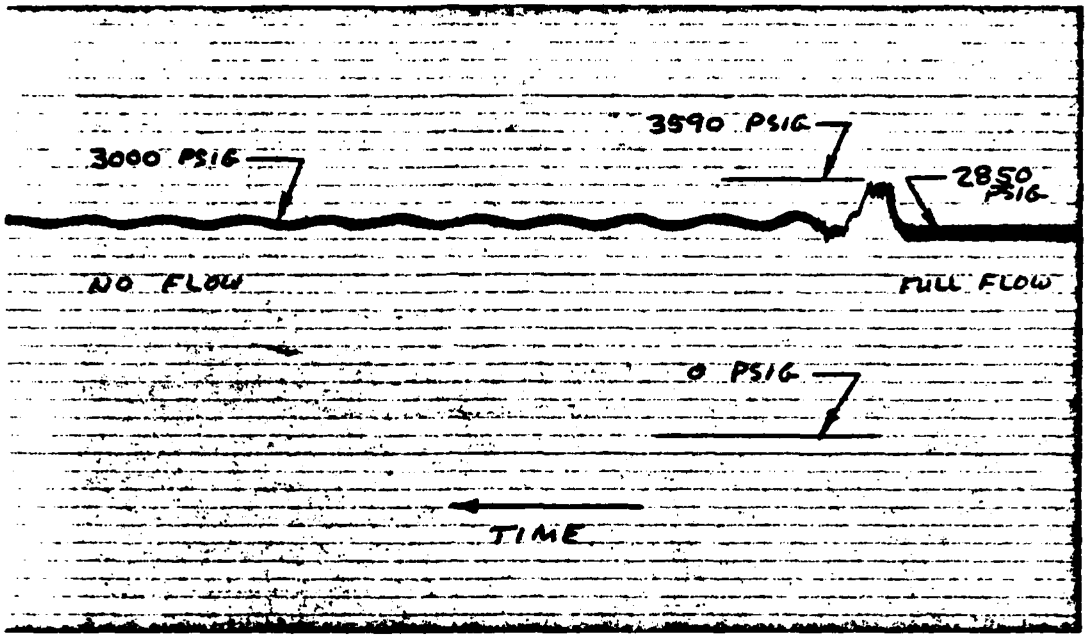


Figure 78a. Pressure transient due to a sudden valve closure while pumping AO-8 fluid into a 250-cu.in. system

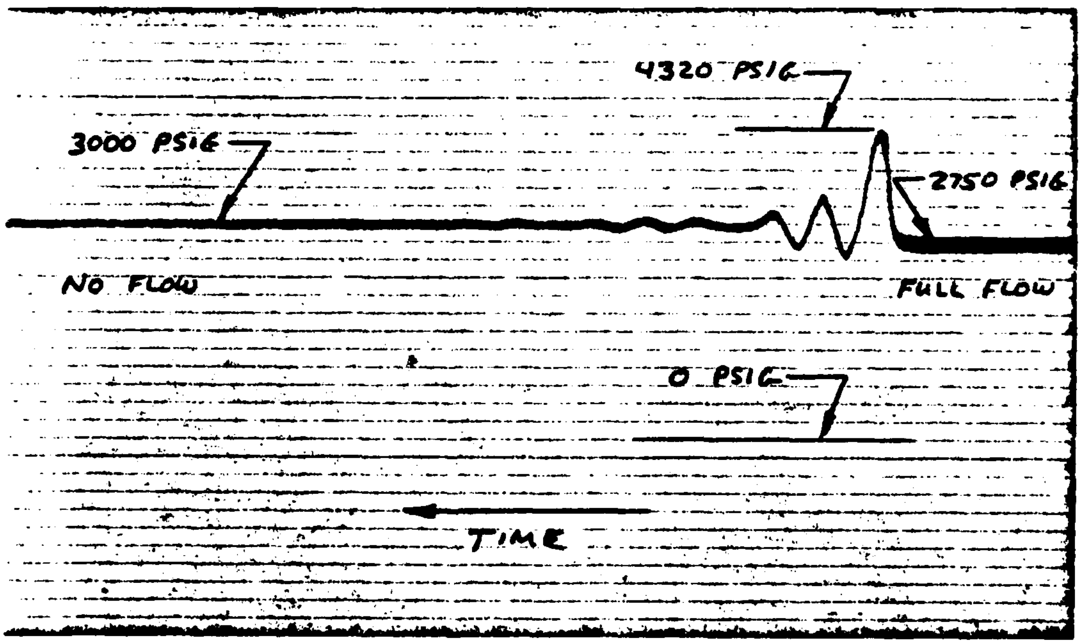


Figure 78b. Pressure transient due to a sudden valve closure while pumping AO-8 fluid into a 50-cu.in. system

Figure 78. Long-term pump transient pressures at 7000 rpm

TABLE 27 TRANSIENT PRESSURE TEST SUMMARY

Pump Speed (rpm)	Inlet Temp. (°F)	Compliance Volume (cu. in.)	Initial Press (psig)	Initial Flow (gpm)	Final Press (psig)	Final Flow (gpm)	Peak Press (psig)
3500	215	50	2850	10.8	3000	0	3890
3500	215	250	2850	10.8	3000	0	3460
7000	215	50	2750	21.3	3000	0	4320
7000	215	250	2850	21.3	3000	0	3590

3.2.2.4.14 Response Time Test

A response time test was performed on 30 and 31 October at pump speeds of 3500, 5250, and 7000 rpm.

Test results are shown in Figures 79 through 81 and are summarized in Table 28.

Response time for all conditions was less than the 0.050 second requirement.

TABLE 28 RESPONSE TIME TEST SUMMARY

Pump Speed (rpm)	Inlet Temp. (°F)	Compliance Volume (cu. in.)	Initial Press (psig)	Initial Flow (gpm)	Final Press (psig)	Final Flow (gpm)	Response Time (sec.)
3500	215	250	2850	10.8	3000	0	0.030
3500	215	250	3000	0	2850	10.8	0.040
5250	215	250	2850	16.0	3000	0	0.040
5250	215	250	3000	0	2850	16.0	0.030
7000	215	250	2850	21.3	3000	0	0.030
7000	215	250	3000	0	2850	21.3	0.030

3.2.2.4.15 Heat Rejection Test

A pump heat rejection test was performed on 30 October 1979.

Test results are plotted and tabulated on Figure 82. Pump heat

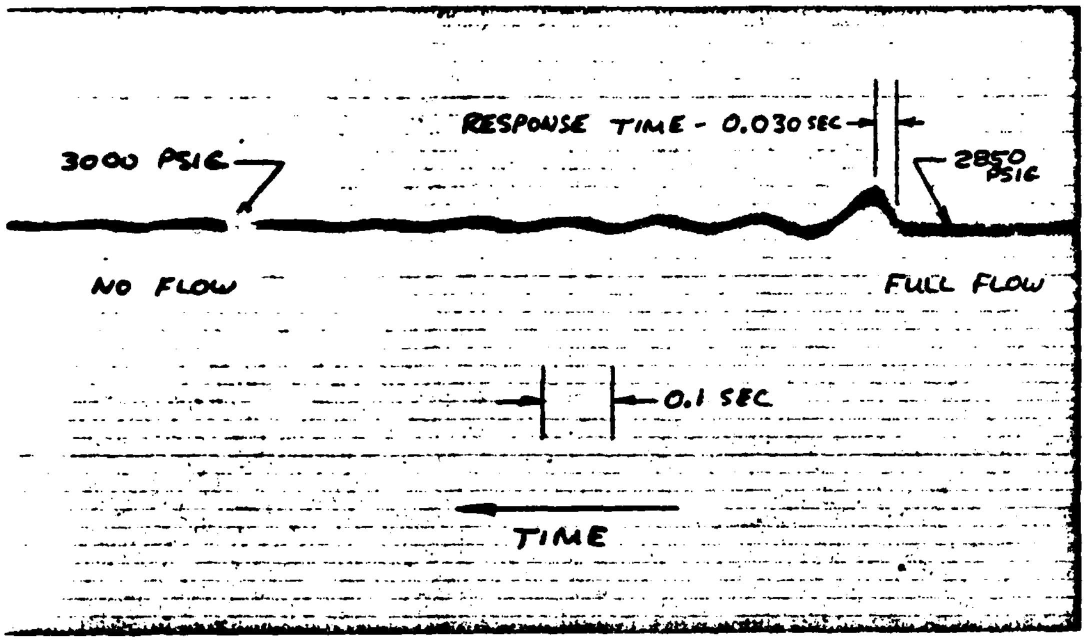


Figure 79a. Pump response following a sudden valve closure while pumping AO-8 fluid into a 250-cu.in. system

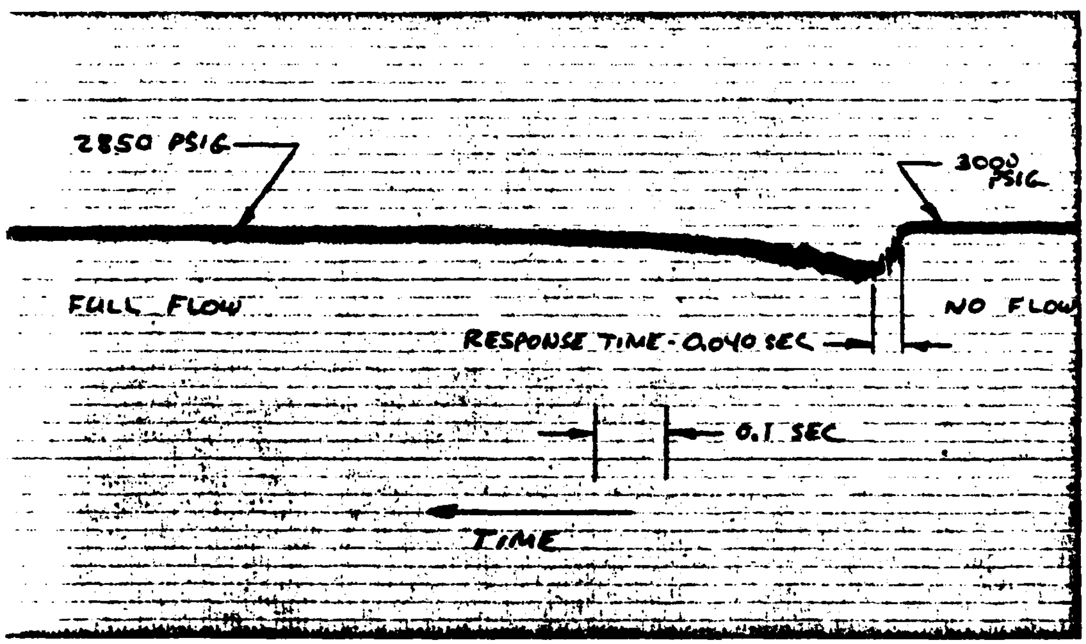


Figure 79b. Pump response following a sudden valve opening while pumping AO-8 fluid into a 250-cu.in. system

Figure 79. Long-term pump response times at 3500 rpm

AD-A118 169 BOEING MILITARY AIRPLANE CO SEATTLE WA F/G 11/8
FIRE RESISTANT AIRCRAFT HYDRAULIC SYSTEM.(U)
JUL 82 E T RAYMOND, D W HULING, R L SHICK F33615-76-C-2064
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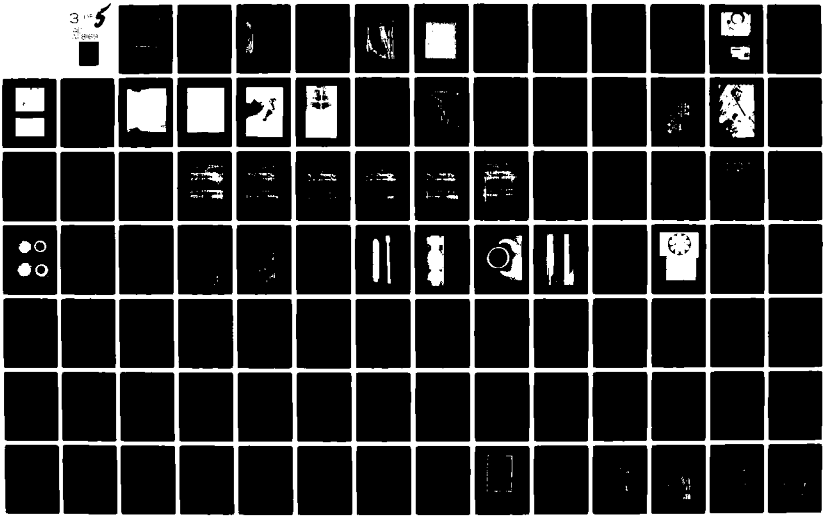


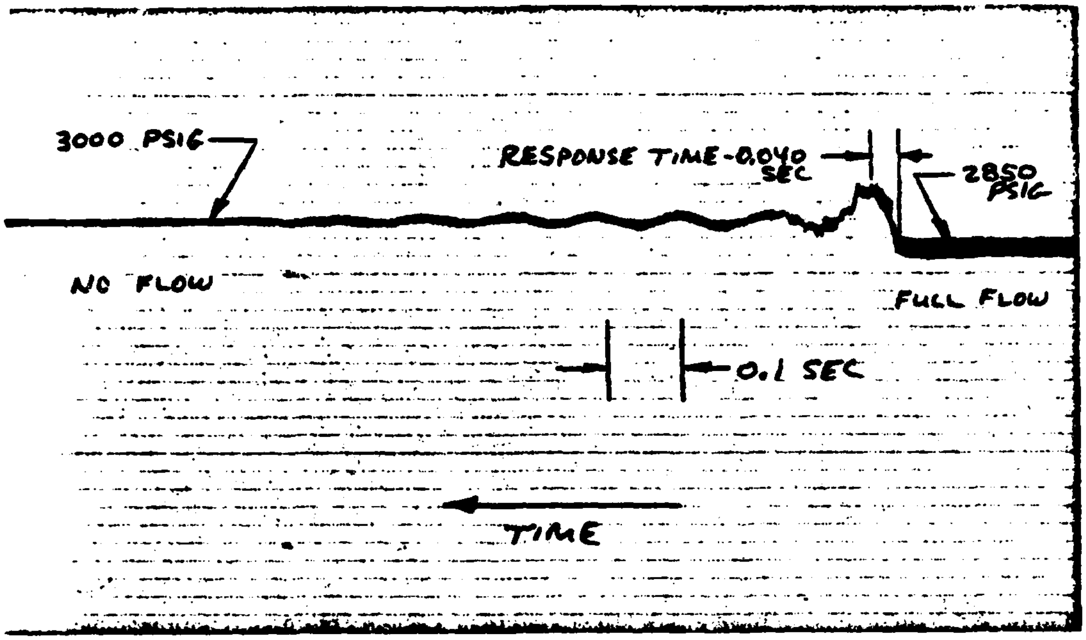


Figure 80a. Pump response following a sudden valve closure while pumping AO-8 fluid into a 250-cu.in. system

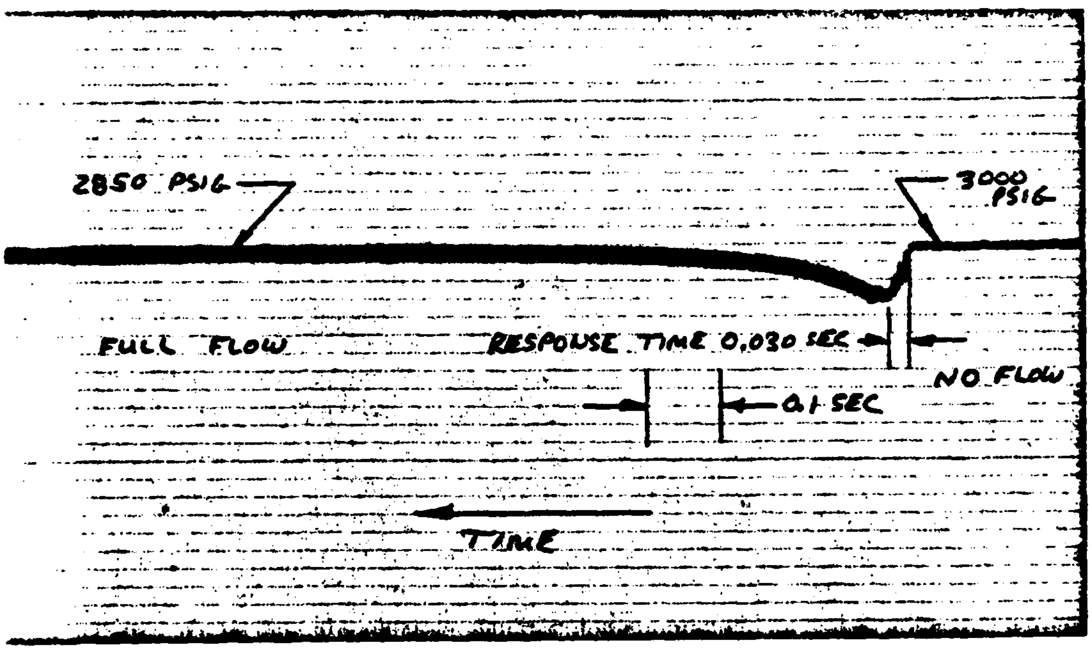


Figure 80b. Pump response following a sudden valve opening while pumping AO-8 fluid into a 250-cu.in. system

Figure 80. Long-term pump response times at 5250 rpm

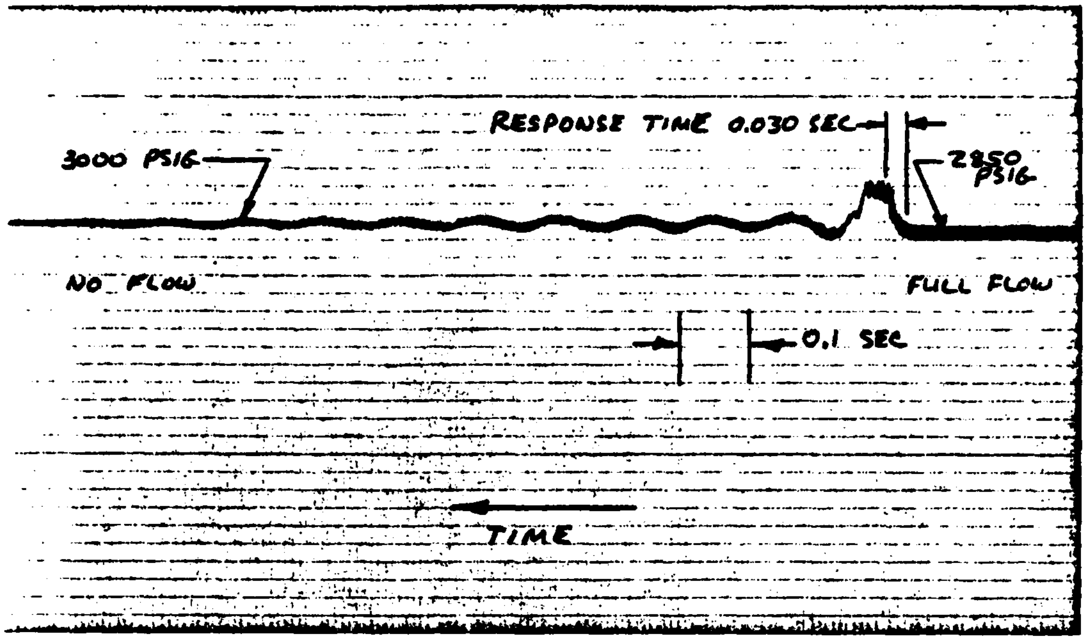


Figure 81a. Pump response following a sudden valve closure while pumping AO-8 fluid into a 250-cu.in. system

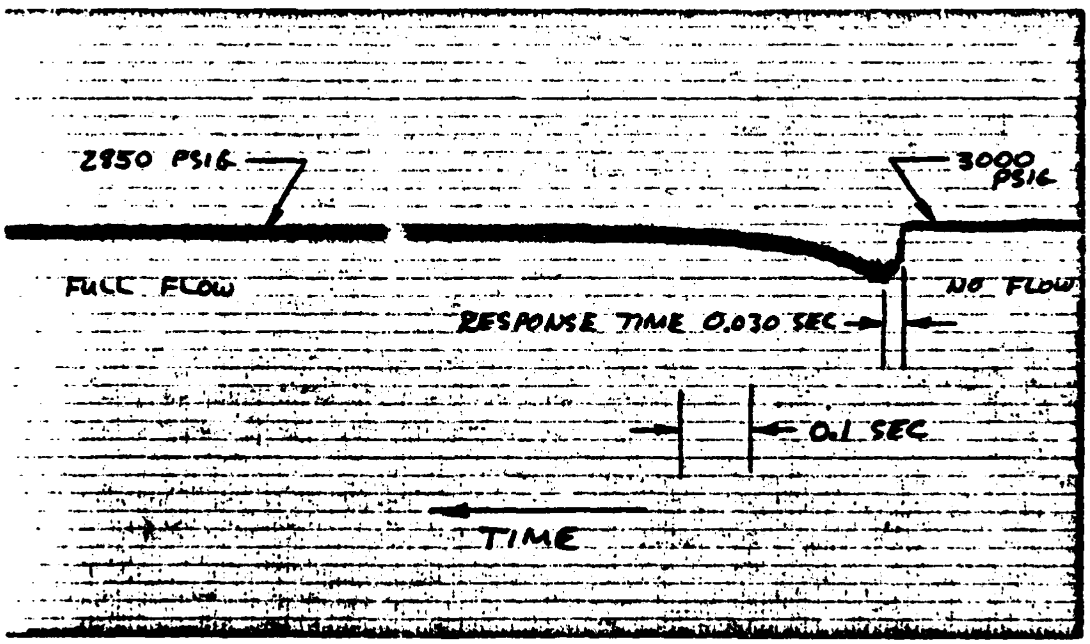
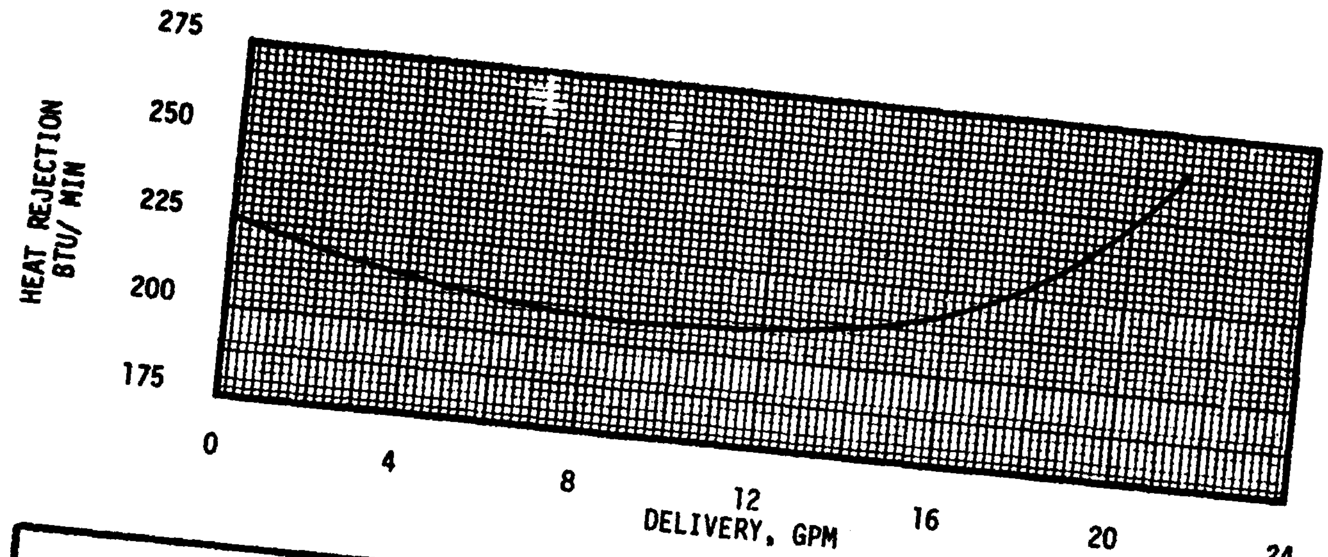


Figure 81b. Pump response following a sudden valve opening while pumping AO-8 fluid into a 250-cu.in. system

Figure 81. Long-term pump response times at 7000 rpm



HEAT REJECTION TEST

Date 30 October 1979 Model No. PV3-075-19 Serial No. MX-337143

Pump Speed (rpm)	PRESSURE Outlet (psig)	PRESSURE Inlet (psig)	Torque (lb-in)	Delivery (gpm)	Inlet Temp. (^oF)	Heat Reject. (BTU/min)
7024	2950	84	48.1	0	208	227.3
7000	2950	86	118.9	4.9	211	212.8
6982	2950	84	196.7	10.1	216	207.9
6960	2940	85	271.8	15.0	207	213.3
6950	2850	84	365.1	21.1	218	263.3

Figure 82. Heat rejection test summary

rejection with the AO-8 fluid was higher in all cases than the Sperry-Vickers qualification test unit with MIL-H-5606 fluid.

3.2.2.4.16 Endurance Test

The planned 750-hour endurance test was started on 31 October 1979 and was terminated prematurely on 21 January 1980 by a pump failure at 713:49 operating hours. Test results were as follows:

3.2.2.4.16.1 Calibration Test

A calibration test at a pump speed of 7000 rpm was performed on 31 October prior to the start of actual endurance pump operation.

Calibration test results are shown in Figures 83 and 84 and compare almost identically to the earlier 7000-rpm data shown in Figures 73 and 74.

3.2.2.4.16.2 Initial Start-Stop Cycles

3.2.2.4.16.2.1 Full-Load Start-Stop Cycles

One hundred thirty full-load start-stop cycles were performed on 1 November 1979.

Pump inlet temperatures ranged from 88 to 210°F. The test cycles are summarized in Table 29.

3.2.2.4.16.2.2 No-Flow Start-Stop Cycles

Sixteen no-flow start-stop cycles were performed on 1 November 1979.

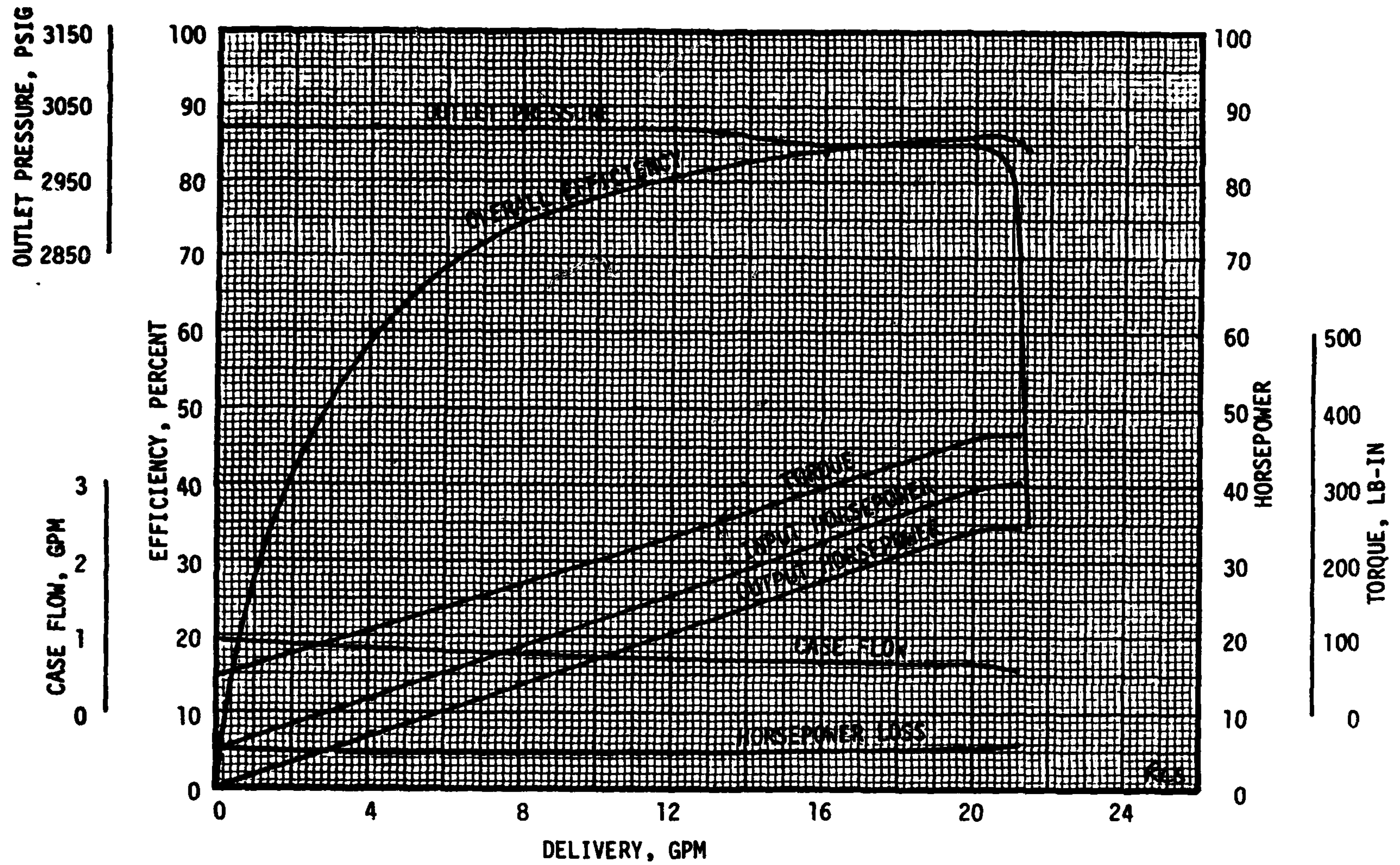
Pump inlet temperatures ranged from 92 to 109°F. The test cycles are summarized in Table 30.

TABLE 30 NO-FLOW START-STOP CYCLES

Cycle	Max. Speed (rpm)	Inlet Temp. °F Start	Inlet Temp. °F Stop	Time, Sec. Start	Time, Sec. Stop	Dischg. Pressure (psig)	Dischg. Flow (gpm)
1	6950	92	99	2.0	22.5	3050	0
5	7111	99	100	2.0	22.5	3050	0
10	7060	105	109	2.0	22.5	3050	0
16	7070	109	113	2.0	22.5	3050	0

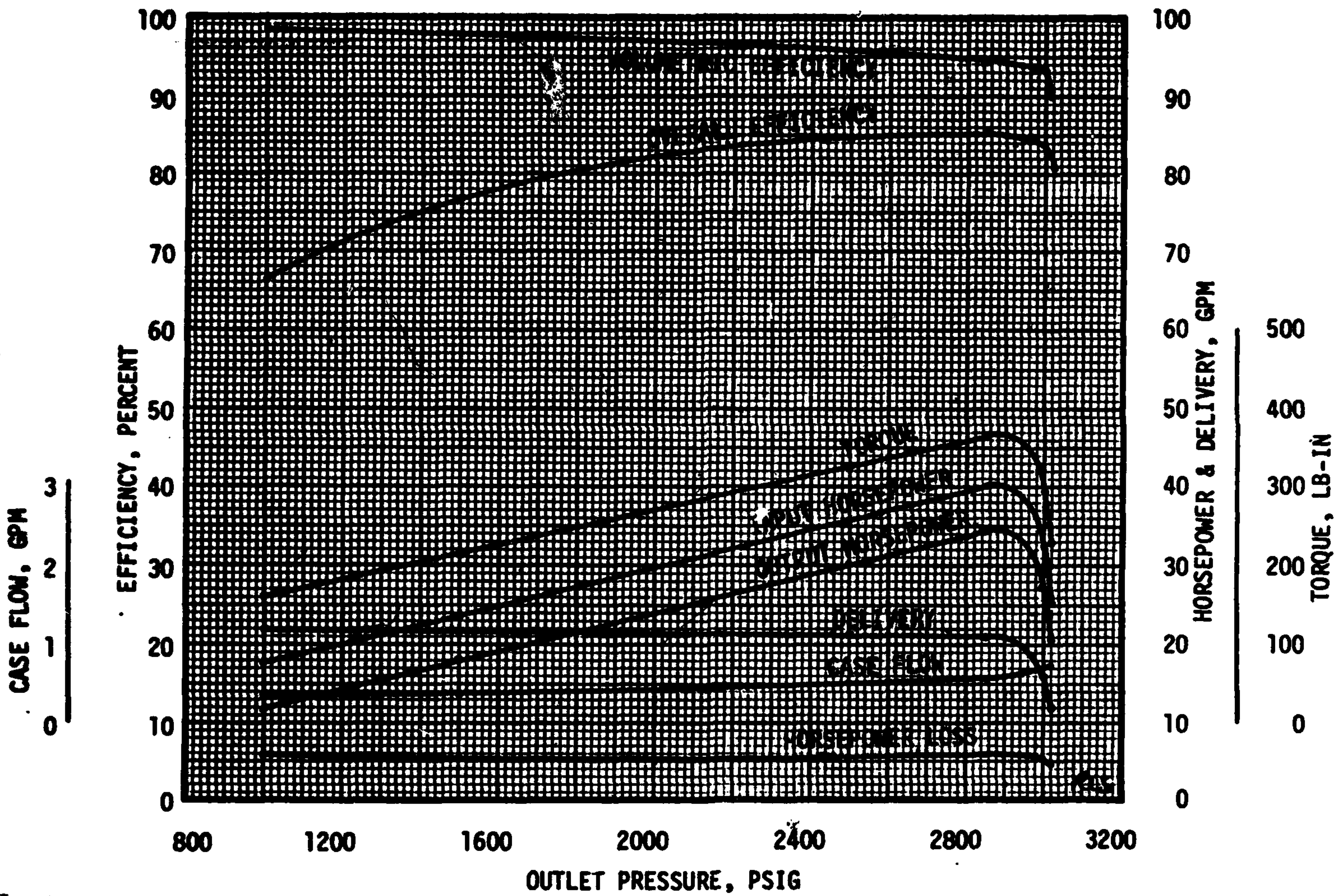
3.2.2.4.16.3 Endurance Test

Actual endurance testing was initiated on 6 November 1979 with the



Fluid: Halocarbon AO-8 Inlet Pressure: 84 psig Temperature: 180 ± 5F

Figure 83. Long-term pump full-flow performance at 7000 rpm



Fluid: Halocarbon AO-8   Inlet Pressure: 84 psig   Temperature: 180 ± 5F

Figure 84. Long-term pump partial-flow performance at 7000 rpm

TABLE 29 FULL-LOAD START-STOP CYCLES

Cycle	Max. Speed (rpm)	Inlet Temp. °F Start	Inlet Temp. °F Stop	Time, Sec. Start	Time, Sec. Stop	Dischg. Pressure (psig)	Dischg. Flow (gpm)
1	6980	113	124	2.0	19.0	2700	22.0
10	6980	127	141	2.0	19.0	2700	21.9
20	6960	151	166	2.0	19.0	2700	21.3
30	7050	175	186	2.0	19.0	2700	21.4
40	6980	194	195	2.0	19.0	2700	21.3
50	6980	210	214	2.0	19.0	2700	21.2
60	6970	108	119	2.0	19.0	2700	21.9
70	6980	141	156	2.0	19.3	2700	21.6
80	6960	166	176	2.0	19.3	2700	21.4
90	6960	187	196	2.0	19.3	2700	21.2
100	6960	184	193	2.0	19.3	2700	21.3
110	6960	160	156	2.0	19.3	2700	21.5
120	6960	98	114	2.0	19.3	2700	21.9
130	6950	88	105	2.0	19.0	2700	22.0

nine-minute cut-off, one-minute full-flow cycle as defined in Table 1 of the test plan.

The cycle times were revised to 4.5 minutes cut-off, 30 seconds full flow on 20 November to reduce temperature fluctuations during cycling and to facilitate stabilizing the test stand temperature.

Table 31 summarizes the actual endurance test conditions and Table 32 summarizes the test schedule and pump operation time.

Pump discharge pressure and flow, case drain flow, and pump fluid inlet temperatures were recorded every 30 minutes during the endurance test. The pressure and flow measurements were stable until the pump failure occurred.

3.2.2.4.16.3.1 Pump Failure

Pump operation was normal during durability cycles 1A through 5A. Pump parameters had been stable and no unusual noise had been generated by the pump. Case drain flow was stable at 0.99 to 1.04 gpm during phase 5A (7000 rpm @ 180°F inlet temperature).

Case flow was fluctuating between 1.25 and 1.34 gpm when the inlet temperature was raised to 215°F for test phase 5B. The pump quickly developed a distressed noise which increased slowly all day before the failure. At 9 hours 14 minutes into phase 5B on 21 January, pump operation became highly erratic and test operation was interrupted. Case flow at the time of failure was oscillating between 1.20 and 1.80 gpm, the case pressure gage was indicating a 30 to 40 psig oscillation with peaks at 150 psig, and the inlet fluid temperature suddenly rose to 260 to 270°F.

The case drain filter bowl was removed and was found contaminated with bronze and steel particles. The decision was made to terminate the long-term pump test.

Total pump operating time at the failure was 713:49 hours.

3.2.2.4.16.4 Remaining Tests

The remaining calibration tests, start-stop cycles, and thermal tests, were cancelled due to the pump failure.

3.2.2.4.16.5 Long-Term Pump Teardown Inspection

The long term test pump was partially disassembled in Wichita on 31 January for a cursory inspection and was taken to Sperry-Vickers on 4 February for detailed inspection where the following conditions were observed.

3.2.2.4.16.5.1 Valve Block

The valve surface was scored and smeared with bronze from the cylinder block (Figure 85).

The bronze transfer may have been due to a lowered valve block surface hardness, perhaps due to a lubrication failure and a resulting overheated condition.

TABLE 31 LONG-TERM PUMP ENDURANCE TEST CONDITIONS

Phase	Speed (rpm)	Duration (Hr:Min)	Nominal Inlet Fluid Temp. °F	CYCLES Flow (gpm)	Pressure (psi)	Duration (min.)	Flow(1) (gpm)	Pressure (psi)	Duration (min.)
1a	3500	150:02	170	0	3000	4.5	10.5	2750	0.5
1b		50:20	215						
2a	5200	150:00	170	0	3000	4.5	15.75	2750	0.5
2b		49:00	215						
3a	6000	75:00	170	0	3000	4.5	18.0	2750	0.5
3b		25:00	215						
4a	6500	75:00	170	0	3000	4.5	19.5	2750	0.5
4b		29:27	215						
5a	7000	37:10	170	0	3000	4.5	21.0	2750	0.5
5b		9:14(2)	215						
6a	7700	0	170						
6b		0	215						

(1) The flow rates shown in Column 8 are the approximate full-flow values for the speeds specified.

(2) Test terminated due to pump failure.

TABLE 32 LONG-TERM PUMP TEST SUMMARY

Test Phase	Start Date	End Date	Phase Time(H:m)	Cum. Time(H:m)
Initial Run-In	8/16/79	8/17/79	4:00	4:00
Pump Teardown	8/22/79	8/22/79	----	4:00
Initial Run-In (cont.)	8/22/79	8/22/79	4:15	8:15
50 Hour Test	8/27/79	9/4/79	49:10	57:25
Pump Rebuild	9/6/79	10/17/79	----	----
Abbreviated Run-In	10/24/79	10/24/79	1:05	58:30
Calibration Test	10/25/79	10/26/79	1:40	60:10
Transient Press/ Response Time	10/30/79	10/31/79	0:35	60:45
Heat Rejection	10/30/79	10/30/79	0:15	61:00
Calibration Test	10/31/79	10/31/79	0:25	61:25
Full Load Start/Stop	11/1/79	11/1/79	0:58	62:23
No Flow Start/Stop	11/1/79	11/1/79	0:13	62:36
Endurance 1A	11/6/79	11/20/79	150:02	212:38
1B	11/20/79	11/27/79	50:20	262:58
2A	11/27/79	12/11/79	150:00	412:58
2B	12/11/79	12/14/79	49:00	461:58
3A	12/14/79	1/2/80	75:00	536:58
3B	1/2/80	1/3/80	25:00	561:58
4A	1/4/80	1/14/80	76:00	637:58
4B	1/14/80	1/16/80	29:27	667:25
5A	1/16/80	1/16/80	37:10	704:35
5B	1/21/80	1/21/80	9:14	713:49

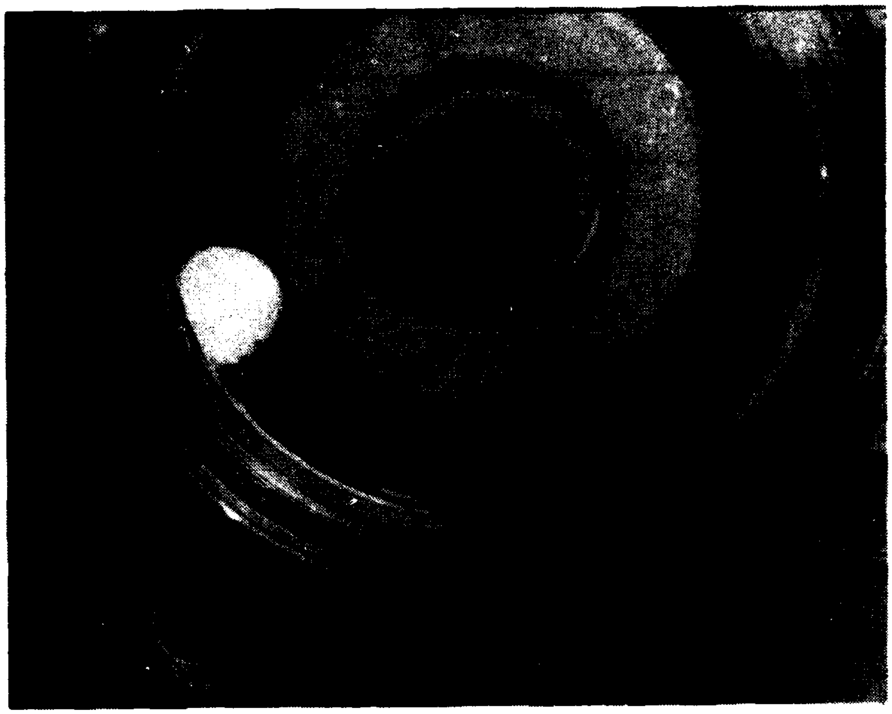


Figure 85. Long-term pump valve block

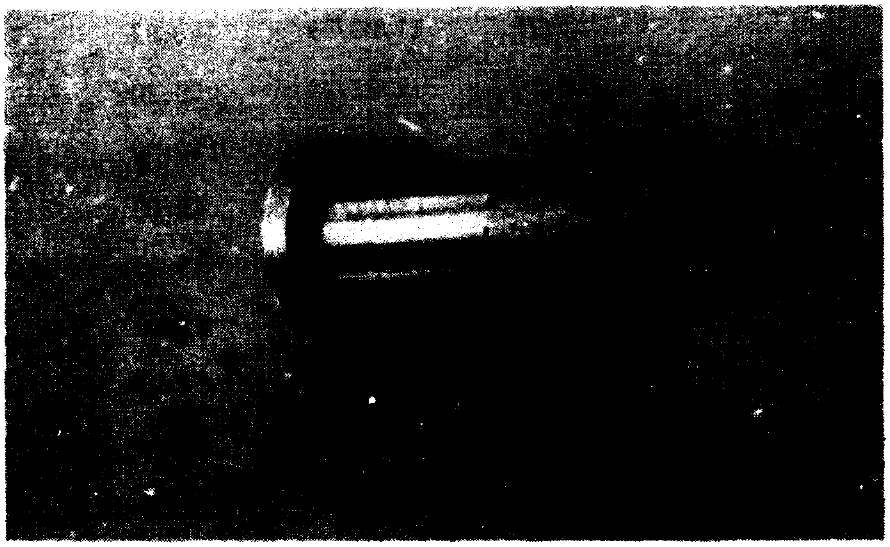


Figure 86. Long-term pump actuator piston

Two cracks approximately 1/8 inch long were found emanating inward from the leakage bleed ports. The cracks were attributed to an overheated surface by an unstable cylinder block. Sperry-Vickers personnel estimated that a surface temperature of approximately 1300°F would be required to redissolve carbides so that quenching would cause cracks.

3.2.2.4.16.5.2 Actuator Piston

The actuator piston was badly galled for the first 3/4 inch of its diameter (Figure 86). The housing bore exhibited no wear. The steel particles from the piston may have been the cause of damage to the Kingsbury bearing pads on the cylinder block.

3.2.2.4.16.5.3 Cylinder Block

The cylinder block was discolored over all external surfaces.

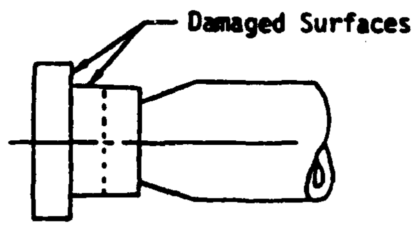
The valve face showed signs of "melting", according to Sperry-Vickers personnel. The Kingsbury pads were badly damaged with three pads exhibiting metal removal to an estimated depth of 0.020 inch (Figure 87).

The cylinder block bores exhibited a coating which reduced the measured bore diameters. The piston-bore measurements indicated a coating of 0.0001 on the bore walls (diameter reduction of 0.0002).

3.2.2.4.16.5.4 Piston-Shoe Subassemblies

The piston barrels were slightly discolored. Erosion of the shoe surfaces (Figure 88) was not obviously greater than after the 50-hour test.

The rear surfaces of the bronze shoes (see sketch) appeared to have imbedded particles of steel in the center of polished "spots" implying that the imbedded particles had upset the softer bronze which had subsequently been worn off. Sperry-Vickers personnel suspected that the silver colored particles were free lead in the bronze which had been exposed by erosion.



Piston barrel diameters had increased by as much as 0.0002 inch. The cause of the coating has not been determined.

Piston to shoe clearances were within tolerance except for shoe number 8 which was 0.0005 too loose.

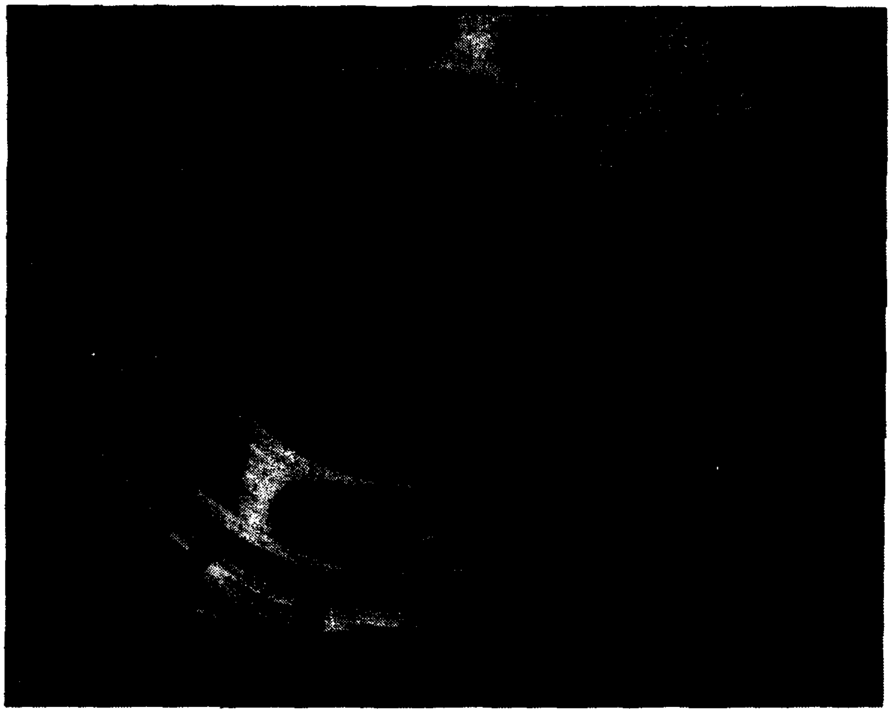


Figure 87. Long-term pump cylinder block

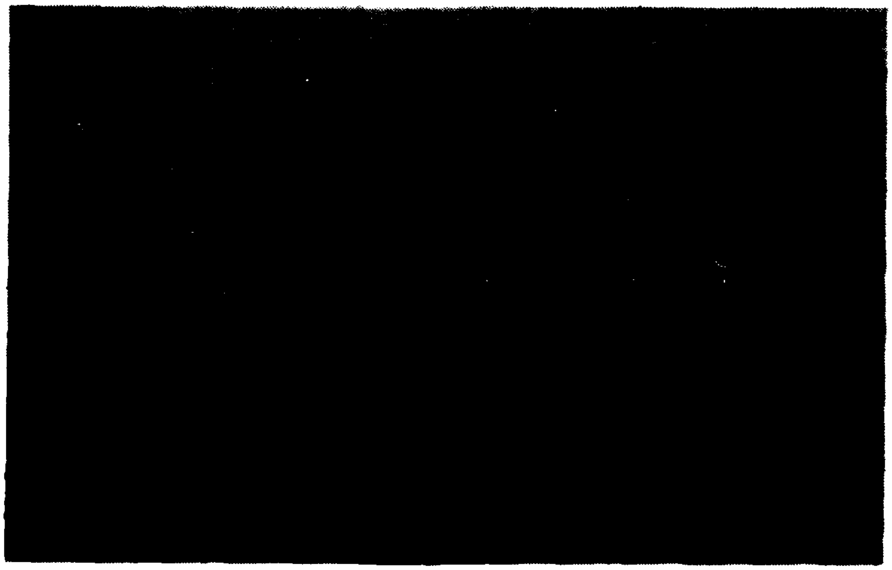


Figure 88. Long-term pump piston shoe

3.2.2.4.16.5.5 Steel Components

Steel components had a dark coating over all surfaces except for wear surfaces. The coating could be partially removed by rubbing, and apparently did not affect function.

3.2.2.4.16.5.6 Retainer Ring

The bronze retainer ring for the piston shoe hold-down plate was pitted or etched on the bottom surface.

3.2.3.4.16.5.7 Compensator Spool

The compensator spool exhibited no erosion damage when viewed under a low-power microscope.

3.2.2.4.16.5.8 Thrust Bearing

The thrust bearing was examined by the Materials Laboratory at Wright-Patterson AFB. Air Force photographs (Figures 89 and 90) of the inner race and several balls indicate that bearing failure had begun.

3.2.2.4.16.5.9 Seals

The PNF pump seals exhibited swelling and softening. Several seals exhibited loss of material due to extrusion.

Figure 91 indicates the type of pump seal extrusion observed during the endurance test. Figure 91 was taken on 14 December 1979 after approximately 400 hours of pump operation.

External pump leakage was impossible to measure due to the rubbery nature of the leakage fluid/seal material. Figure 92 shows the leakage residue material from under the pump and pump/test stand interfaces during a one-week period from 10 January through 17 January 1980.

3.2.2.4.16.6 Sperry-Vickers Teardown Inspection Report

The Sperry-Vickers teardown inspection report is included as Appendix I. The discrepancy between test hours on the pump is due to an error in totaling the initial test hours.

3.2.2.4.16.7 Fluid Condition

3.2.2.4.16.7.1 Fluid Description

The test fluid was Halocarbon AO-8 fluid with the anti-wear lubricity improver Molyvan A added.

3.2.2.4.16.7.2 Fluid Analysis

All filters utilized in the long term pump test stand were 5-micron absolute rated units provided by Aircraft Porous Media. The standard housing seals had been replaced with PNF seals and the filter elements were fabricated with epoxy which was compatible with the AO-8 fluid.

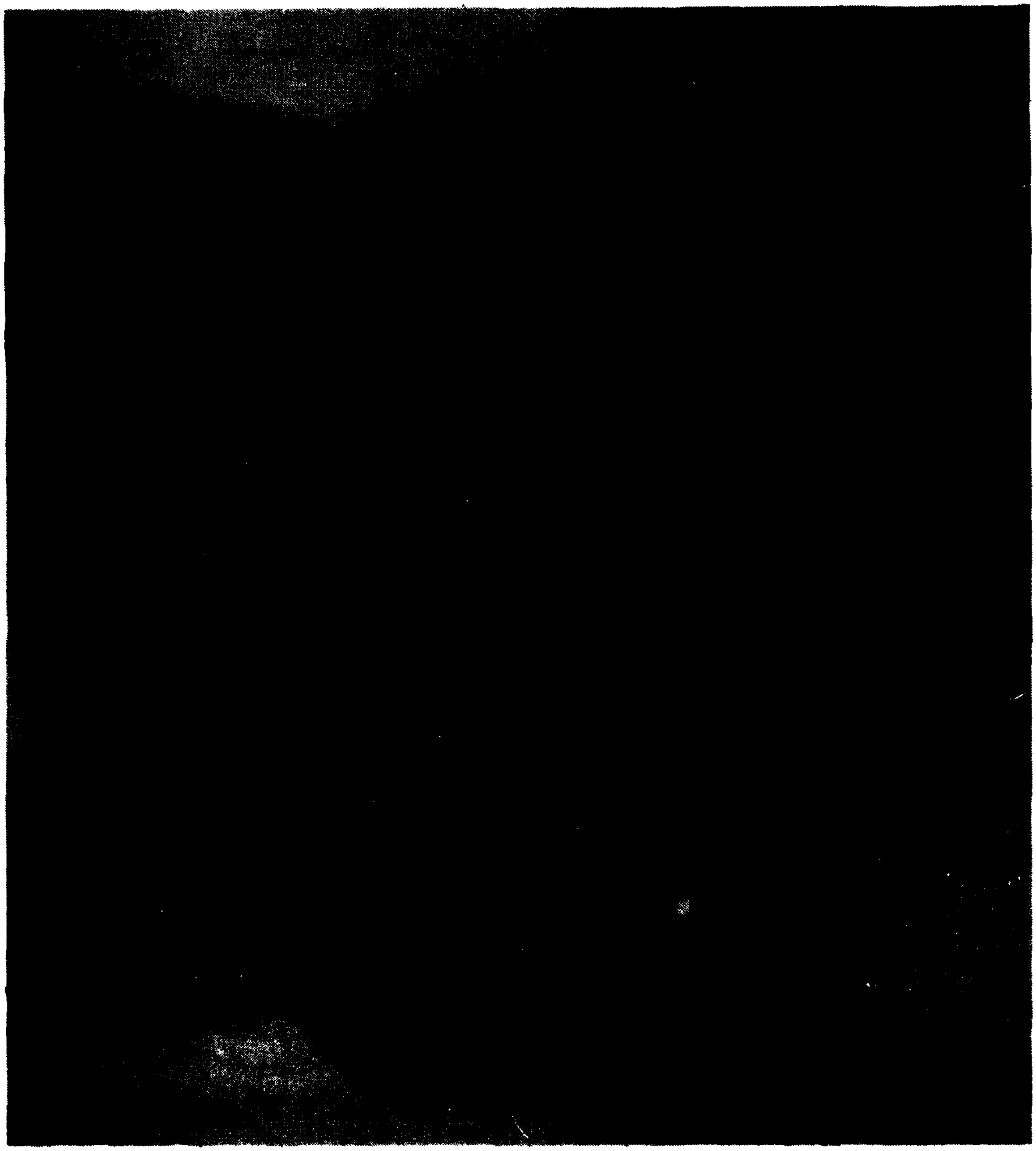


Figure 89. Long-term pump thrust bearing inner race

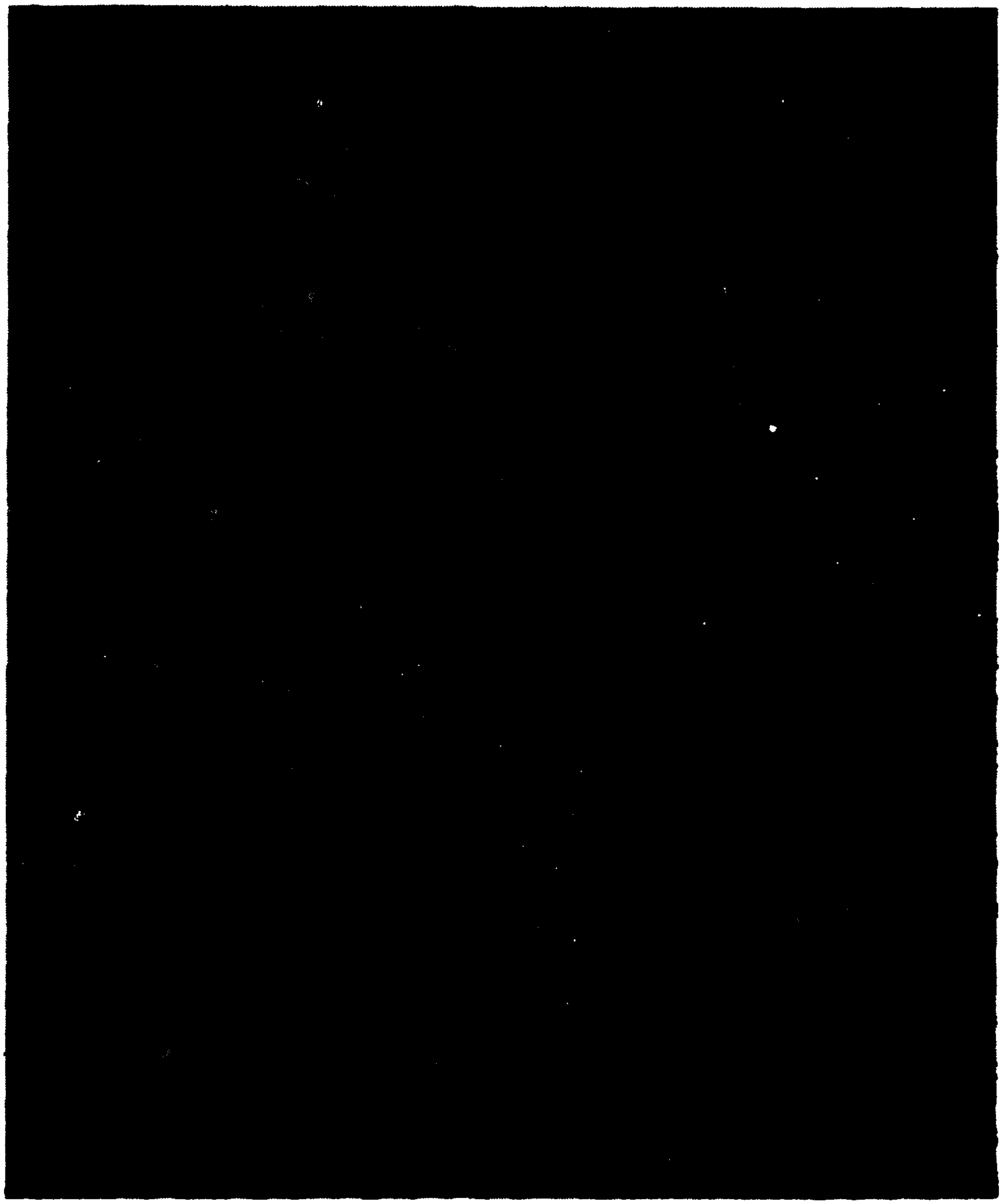


Figure 90. Long-term pump thrust bearing balls (typical)

Figure 91. Long-term pump seal extrusion

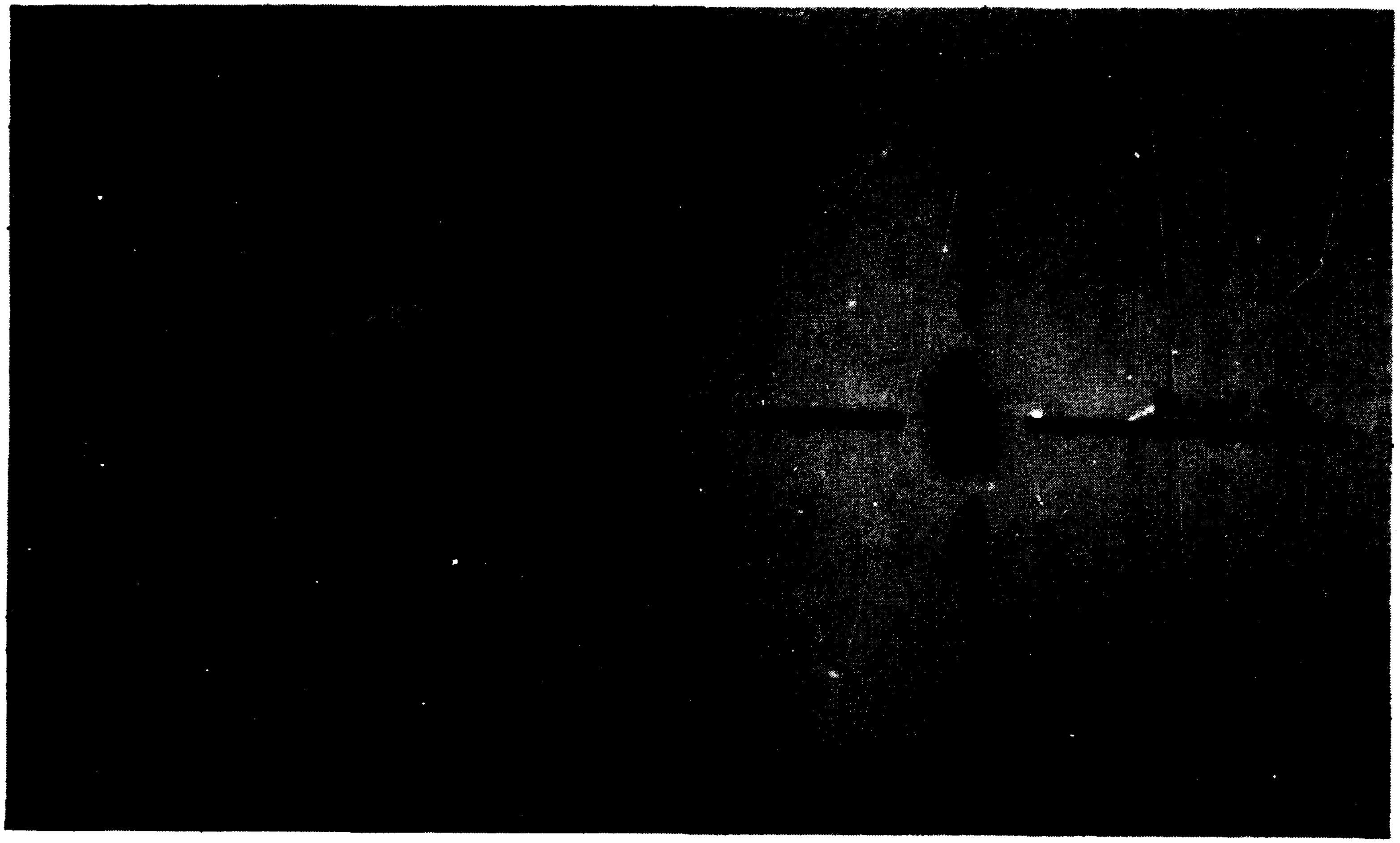


Figure 92. Long-term pump seal leakage residue

Fluid samples were taken upstream of the discharge and case drain filters periodically during the pump test. Particle counts performed on the fluid samples by Scientific Laboratory Services of the Pall Corporation were utilized to assign contamination classes per NAS 1638, Reference 9, to the fluid samples. The results are tabulated in Table 33.

TABLE 33 LONG-TERM PUMP TEST STAND FLUID SAMPLES

Date	Approx. Pump Hours	NAS-1638 Class Case Drain	NAS-1638 Class Discharge
15 Nov. 1979	168	---	5
16 Nov. 1979	183	7	---
6 Dec. 1979	384	6	6
17 Dec. 1979	480	8	7
4 Jan. 1980	575	7	7
17 Jan. 1980	666	12+	10

As shown in Figure 93, the pump case drain effluent worsened near the end of testing. The pump discharge effluent cleanliness appears to be a function of pump speed, with wear occurring at all speeds. Obviously a 20 to 25 percent decrease in the rated speed of the pump will not produce an infinite life. Additional testing would be required to estimate pump life as a function of speed.

3.2.2.4.17 Cold-Start Test

As mentioned previously, in 3.2.2.2.2, the cold-start test was performed with the refurbished fifty-hour test pump (S/N MX-319687).

Five starts were made at minimum outlet pressure, two with the outlet blocked, and two with a 2700-psi relief valve across the outlet.

Test data is summarized in Table 34. Outlet flow or pressure was developed in all starts. The time to reach stable pump operation varied between 4 and 15 seconds, with considerable noise and discharge pressure fluctuation occurring in all cases until stable pump operation was achieved. Typical discharge pressures fluctuated between 2500 psi and 3000 psi.

9. NAS 1638, National Aerospace Standard, Cleanliness Requirements of Parts Used in Hydraulic Systems, Aerospace Industries Association of America, Inc., Washington, D.C., January 1964.

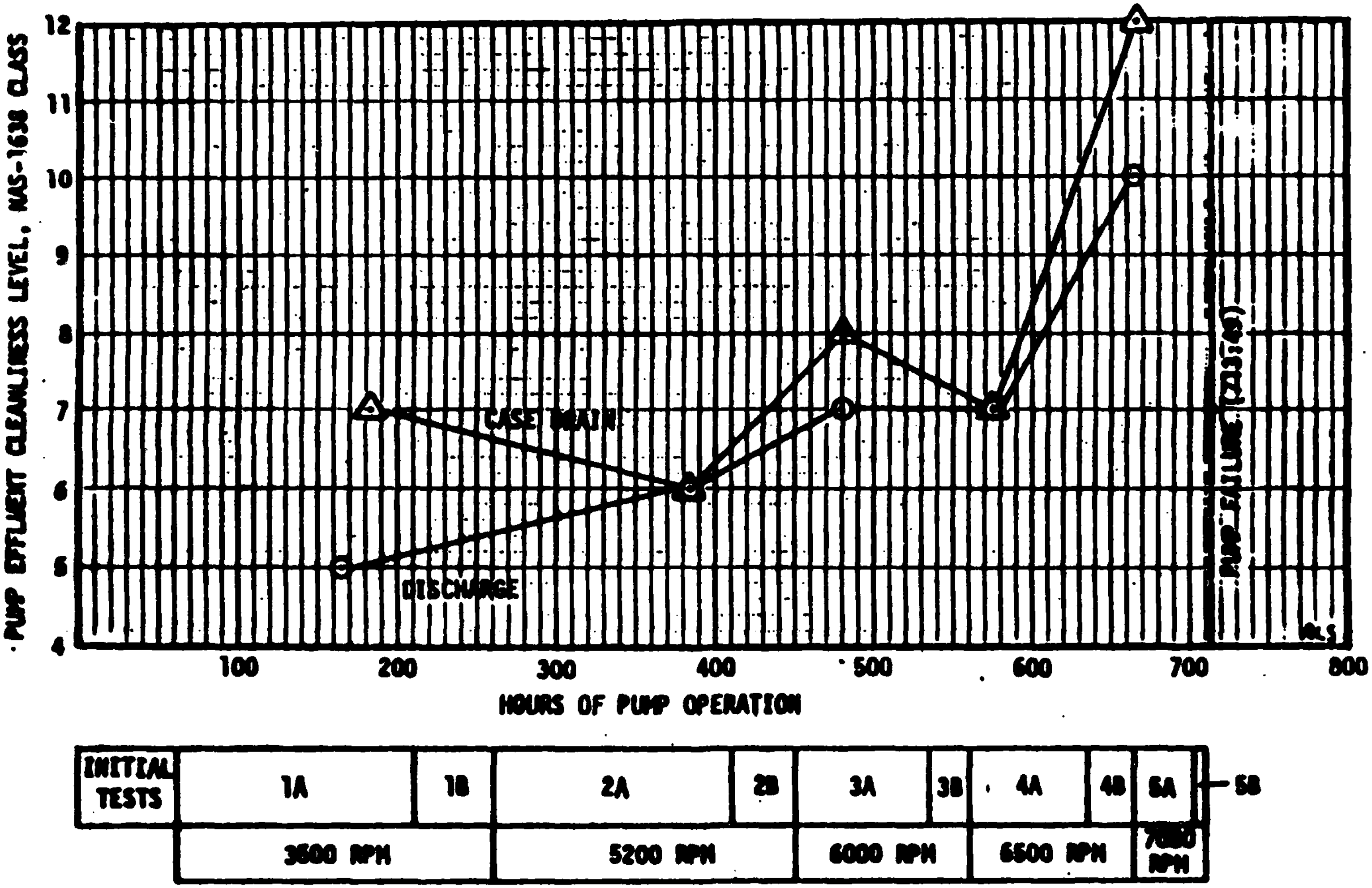


Figure 93. Long-term pump effluent cleanliness levels

TABLE 34 PUMP COLD-START TEST SUMMARY

Start No.	Pump Outlet	Initial Temperature: Pump Case (°F)	Initial Temperature: Reservoir (°F)	Time to Flow (sec.)	Outlet Flow Rate (gpm)	Outlet Press: Peak (psig)	Outlet Press: Stabilized (psig)	Date
1		-67.6	-64.5	10	21.4	3000	400	26 Mar '80
2	Minimum	-65.8	-63.4	8	21.5	2500 - 3100 OSC	400	26 Mar '80
3	Pressure	-64.0	-63.5	7	21.2	2500 - 3000 OSC	350	26 Mar '80
4		-63.0	-70.3	12	21.6	2500 - 3000 OSC	400	27 Mar '80
5		-65.0	-66.5	8	21.6	2500 - 3000 OSC	400	27 Mar '80
6		-65.0	-66.3	5	0	3300	3300	27 Mar '80
7	Blocked	-65.2	-68.1	4	0	3250	3250	28 Mar '80
8	2700 psi	-62.3	-62.9	10	21.0	3300	2700	28 Mar '80
9	Relief Valve	-62.6	-63.4	15	23.8	3300	2700	28 Mar '80

Pump speed - 0-7000 RPM, 2 sec.
Inlet Press 85 psig

Sperry-Vickers Pump
M/N PV3-075-19
S/N MX-319687

3.2.2.5 Long-Term Pump Test Conclusions

The completion of nearly 714 hours of rigorous pump operation can be considered an unqualified success since the AO-8 fluid is still in the preliminary stages of development.

The corrosion of bronze alloys and the lack of proper lubrication continue to be significant problems with the AO-8 fluid.

It appears that short-term operation at low speeds causes a loosely adhered scale to form on bronze components. High-speed operation apparently scrubs the scale off of the bronze surfaces and discolors the fluid. Long-term corrosion effects must still be investigated.

The lack of proper lubrication has plagued the AO-8 fluid since testing has begun. The condition of the two pumps after testing with the AO-8 fluid with the Molyvan-A antiwear additive was better than the post-test condition of the 50-hour test pump. The consistent cylinder-block/valve-block problems and thrust bearing problems indicate that additional fluid development work is required.

The speed ratings of current pump designs operating with MIL-H-5606 fluid may be too high to obtain satisfactory pump wear life with the AO-8 fluid. Insufficient testing has been performed to adequately determine the optimum rated speed for the PV3-075 pump operating the AO-8 fluid.

3.3 HYDRAULIC SERVOACTUATOR TESTS

These tests were run to determine the performance and life of a typical hydraulic servoactuator, operating with the selected nonflammable fluid, for compariason with results obtained during qualification testing with MIL-H-5606 fluid. The test fluid was Halocarbon AO-8, Batch No. 62979, with the lubricity and anti-wear additive Molyvan A (molybdenum oxysulphide dithiocarbamate) added. Again, as in the pump tests, Firestone PNF Compound 200R-211658 elastomeric O-ring seals molded by Nichols Engineering were used.

3.3.1 Test Item

The servoactuator utilized for this test was a B-52 elevator power control unit (part number 27830-4) manufactured by the Nuclear Valve Division of Borg-Warner, formerly Weston Hydraulics Division of Borg-Warner. The functional diagram is shown in Figure 94.

The power control actuator consists of a dual-tandem hydraulic actuator with integrated input servomechanisms and internal position feedback and summing linkage. The servoactuator is capable of operating in the mechanical mode with mechanical input commands to the pilot input lever, and in the combined mode with series summing of mechanical input commands and electrical input commands to either or both of the two flow control valves. The flow from each flow control valve is directed to an auxiliary actuator which drives the summing linkage. A position transducer is attached to each auxiliary actuator to close the control loop through a valve drive amplifier and demodulator. Two shutoff valves are provided to allow selection of operating modes. Normal operation is in the combined mode utilizing only one electrical channel.

Personnel from the Air Force Air Logistic Center in Sacramento, California disassembled, cleaned, and reassembled the servoactuator at Boeing-Wichita on August 22, 1979 prior to testing in the AO-8 fluid. PNF seals were installed throughout the actuator except in the servovalves and the seal plates used to mount the servovalves and solenoid valves. Also installed at that time was a modified control valve having widened metering slots for equivalent output performance with the more dense AO-8 fluid.

3.3.2 Test Setups

3.3.2.1 Performance/Endurance Test Setup

The test setup utilized for all performance and endurance tests is shown in Figure 95 and is depicted schematically in Figure 96. Equipment utilized in the test setup is tabulated in Table 35.

The servoactuator was installed in the ECP 1195 qualification test fixture at Boeing-Wichita for all performance and endurance tests.

The pump utilized for the servoactuator test was the Sperry-Vickers PV3-075-19 (50-Hour Test) pump which had been refurbished by Sperry-Vickers prior to this test (see Section 3.3.4.11).

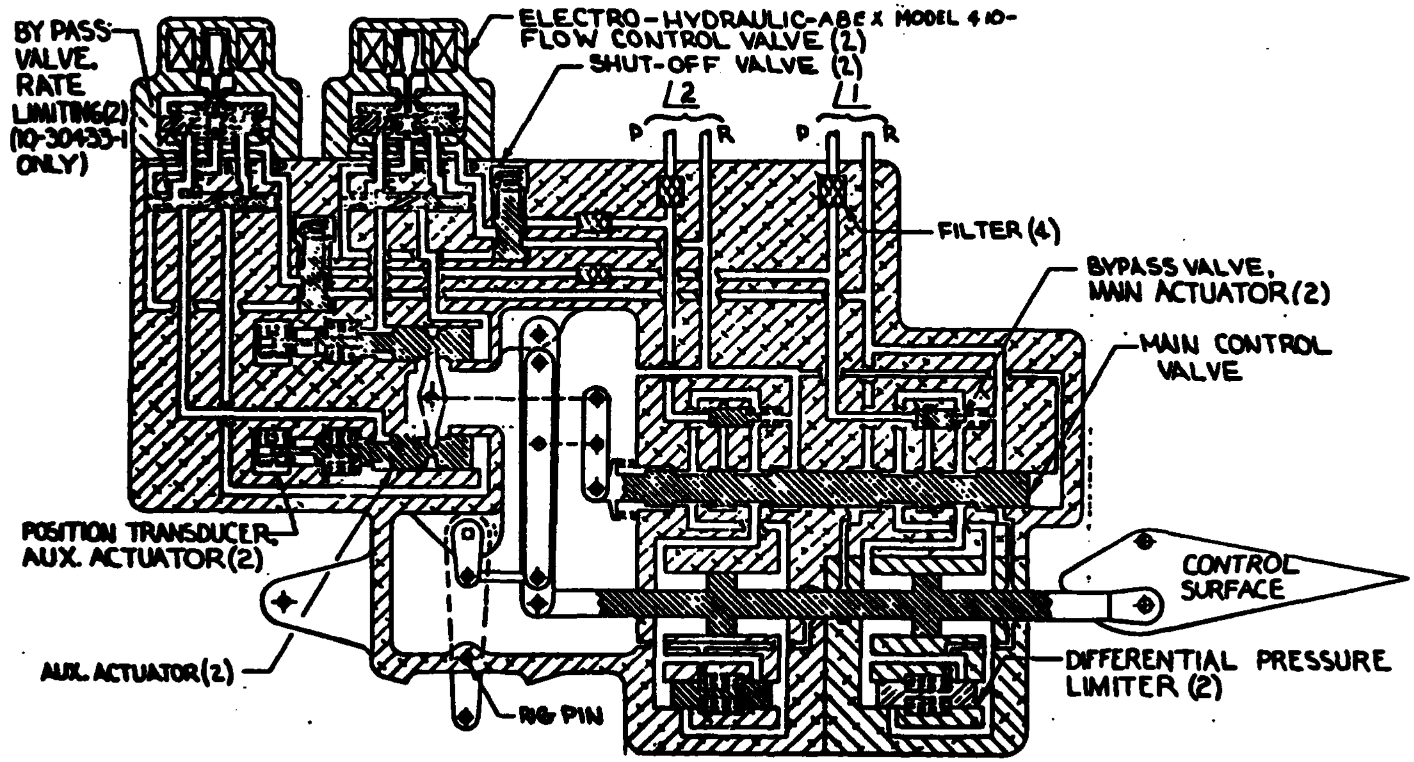


Figure 94. Servoactuator functional diagram

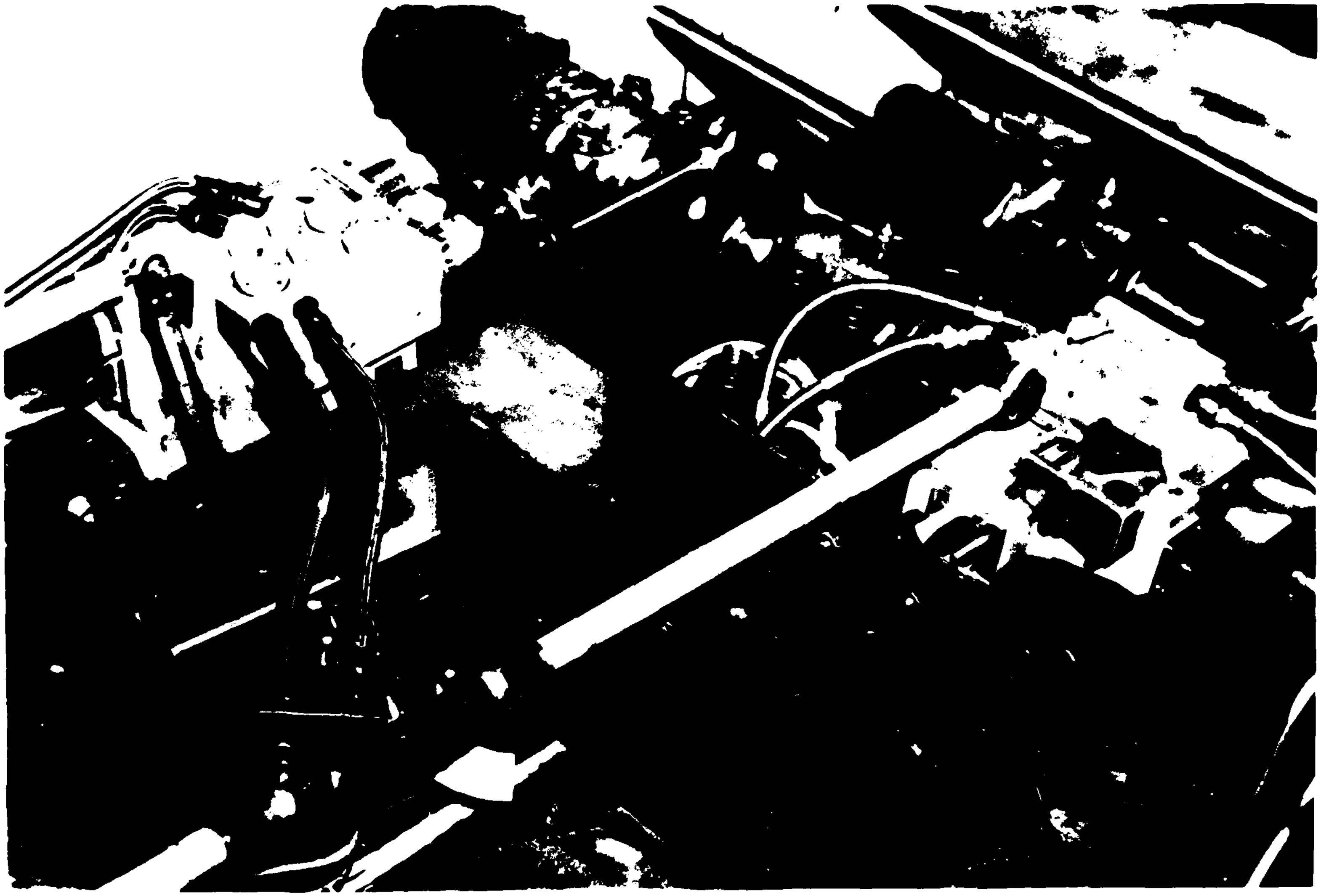
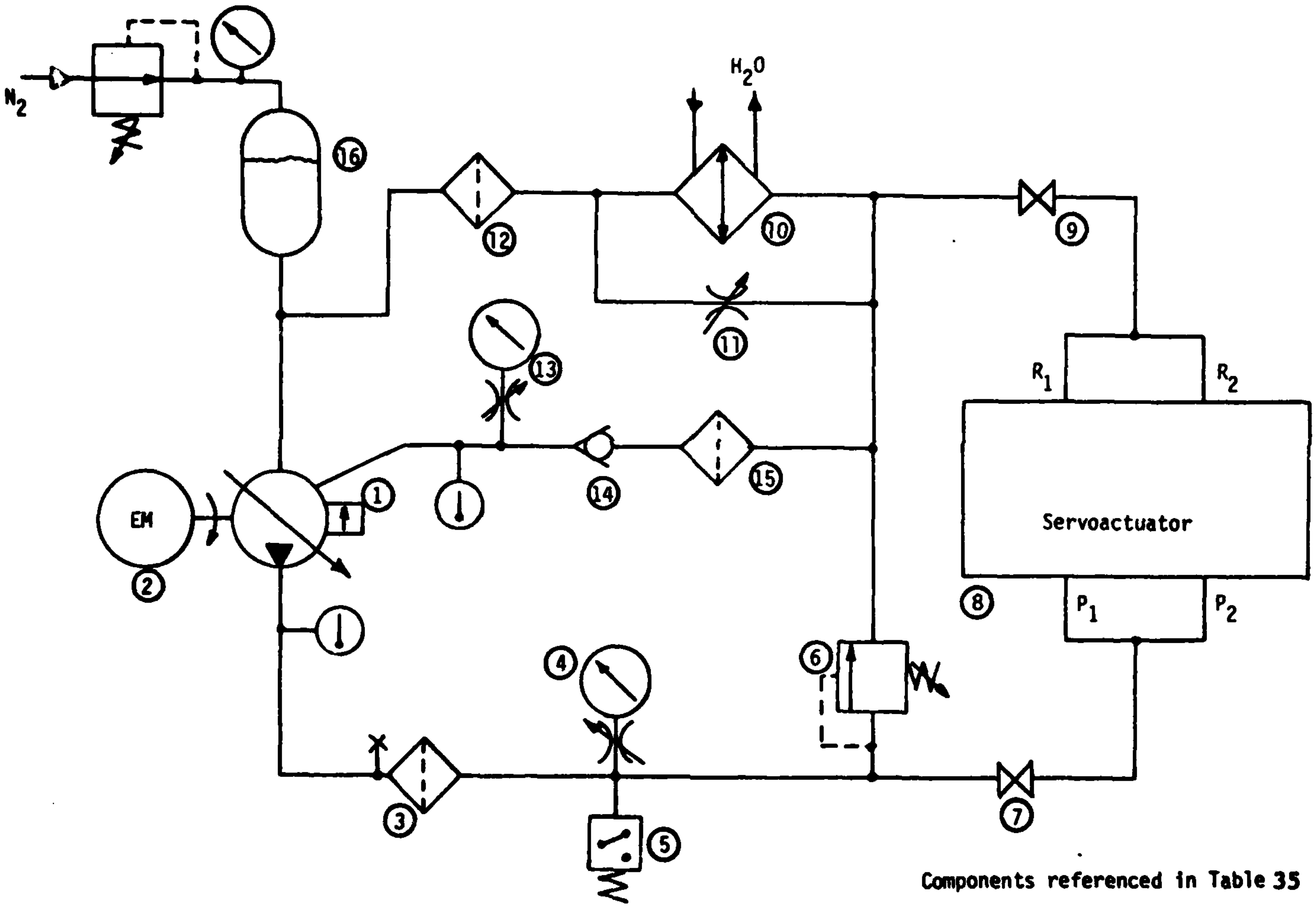


Figure 95. Servoactuator loading system



Components referenced in Table 35

Figure 96. Servoactuator test setup

TABLE 35 SERVOACTUATOR TEST EQUIPMENT LIST

1. Pump, Sperry-Vickers, PV3-075-19, S/N MX-319687
2. Varidrive, U.S. Motors, VEU-GHDT, 15 hp, S/N 280735
3. Discharge Filter Assy, Aircraft Porous Media, AD-3255-16Y77, S/N 0528
4. Discharge Pressure Gage, Ashcroft, 5000 psig, 50-psi increments, S/N RL4060
5. Discharge Pressure Switch, Hydra-Electric Co., 15015-1, S/N 480
6. Relief Valve, Mission Valve, 151102-2120, AND-12 x AND-16, S/N 1916-3
7. Shutoff Valve, Parker Hannifin, MV-1230-S
8. Test Servoactuator, Weston Hydraulics, 27830-4, S/N 435468
9. Shutoff Valve, Parker Hannifin, MV-830-S
10. Heat Exchanger, Vickers Inc.
11. Bypass Valve, Parker Hannifin, MV-1230-S
12. Return Filter Assy, Aircraft Porous Media, AD-3258-16D-117, S/N 9680-1770
13. Case Drain Pressure Gage, Ashcroft, 400 psi, 2-psi increments, S/N 13-2387
14. Check Valve, Teledyne Republic, MS24593-6
15. Case Drain Filter Assy, Aircraft Porous Media, AC-3258-68Y15, S/N 0042
16. Reservoir, Boeing Co., 35-3297
17. High-Pressure Reservoir (for servoactuator cold tests), Parker Aircraft, 2660359, 200 cu.in. accumulator (piston removed)

3.3.2.2 Cold Test Setup

Functional tests of the actuator at a -65F ambient temperature required alteration of the performance/endurance test setup to allow refrigeration of the servoactuator and an adequate quantity of fluid. Also, a high-pressure reservoir was installed in the servoactuator pressure line as depicted in Figure 97, and an environmental chamber was built around the servoactuator and high-pressure reservoir. The high-pressure reservoir is described in Table 35 (Item 17).

3.3.3 Test Performance Period

The servoactuator test was performed over the period of 27 August 1979 to 3 March 1980 in compliance with the Servoactuator Test Plan, Appendix J. Since the servoactuator and long-term pump tests were planned to run concurrently to minimize program costs, servoactuator testing was interrupted from 6 September 1979 to 17 October 1979 during repair of the long-term test pump.

3.3.4 Test Results

3.3.4.1 Pre-Test Teardown Inspection

The condition of the servoactuator at the initial teardown was generally excellent except for a rough condition of the Number 2 cylinder bore in the aluminum housing. The Number 2 piston teflon cap strip was nearly worn through due to the bore condition, but all other internal components showed no wear and were well preserved with no corrosion or discoloration.

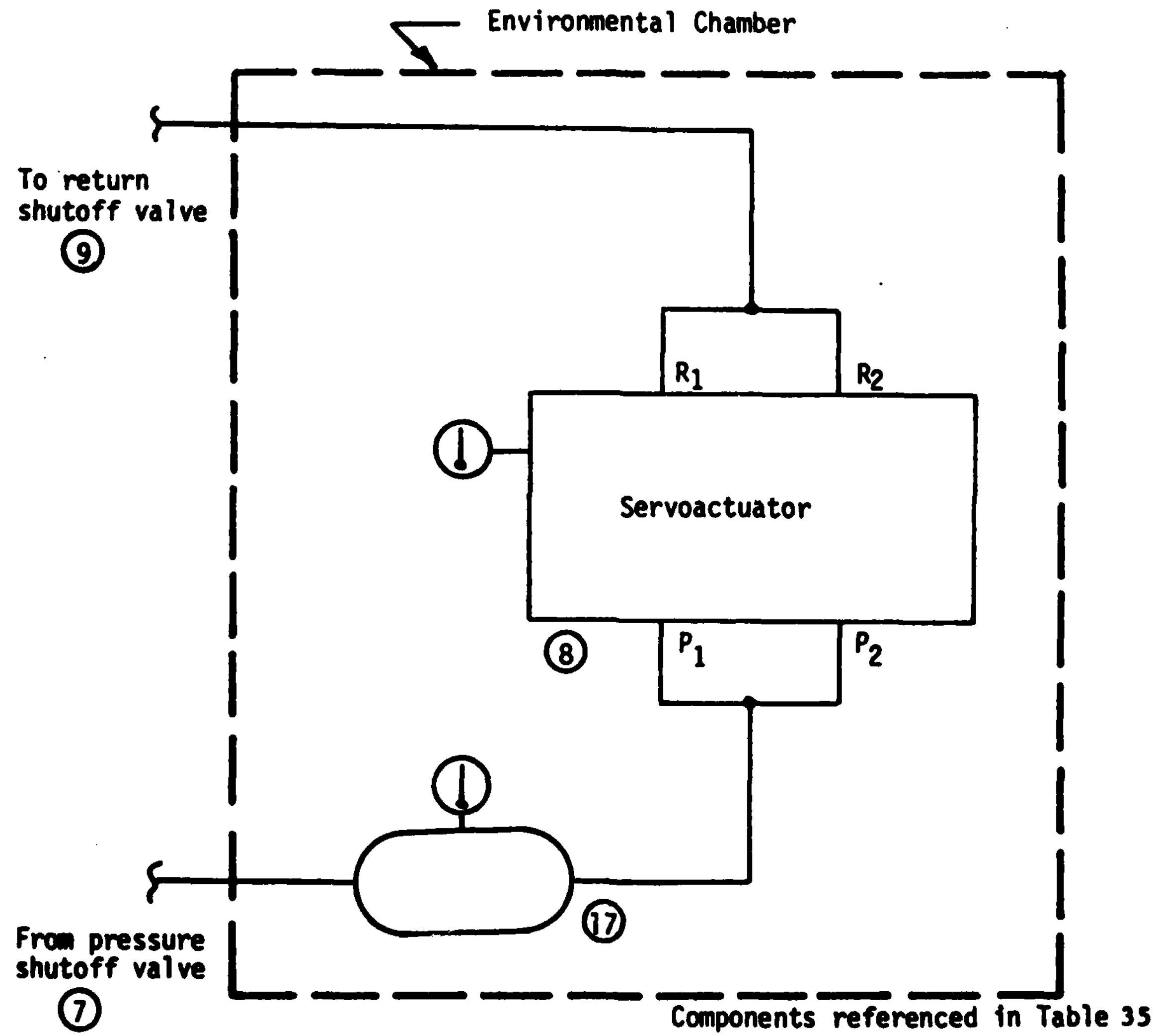


Figure 97. Servoactuator cold-test setup

The servoactuator was reassembled with the modified control valve, using PNF packings in all applications except the internal servovalve seals, seal plates for the servovalves and solenoid valves, and electrical connector seals not in contact with the fluid. All teflon cap strips, piston seals, and foot seals were replaced with new seals from a servoactuator overhaul kit.

3.3.4.2 Velocity Test

A no-load velocity test was performed on 30 August 1979, and the test results were as follows:

	Velocity (in/sec) Extend	Velocity (in/sec) Retract	Fluid Temp (F)
Measured Value	6.95	6.67	105
Specified Requirement	6.25	6.25	100 ± 20

Previous calculations had indicated that the control valve metering slot width should be increased from 0.062 inch to 0.092 in order to maintain the desired 80 degree per second no-load actuation rate with the more dense AO-8 fluid.

The velocity test confirmed that the control valve modification had achieved the desired no-load rate (6.25 in/sec = 80 deg/sec).

3.3.4.3 Dynamic Response Tests

A dynamic response test was performed on both servoactuator channels on three occasions. The first test was performed at a nominal fluid temperature of 100F on 31 August 1979 with the servoactuator in a newly reassembled condition. The second test was performed on 19 February 1980 after the 250,000-cycle endurance test, again with a nominal fuid temperature of 100F. The third test was performed on 20 February 1980 with a nominal fluid temperature of 250F. A fourth dynamic response test originally scheduled to be performed at -65F was deleted because there is no specification requirement for performance at -65f.

Bode plots of the dynamic response tests are included as Figures 98 through 103. The dynamic response, although generally improved after the endurance test (Figures 102 and 103), was not as good as predicted by a Boeing-Wichita analog computer simulation in 1978. The analog computer simulation indicated that the performance with AO-8 fluid at 275F with 0.092-inch metering slots should essentially match the performance with MIL-H-5606 at 160F with .062-inch metering slots.

The probable causes of the reduced performance is the excessive seal friction present throughout most of the endurance testing and the fluid temperature discrepancy between the tests (100-110F) and the analog simulation (275F).

The improvement in dynamic response after the endurance test was due to decreased seal friction and the much higher fluid temperatures (245F). The reduction in seal friction was confirmed by the excessive seal wear discovered during the post-test teardown inspection.

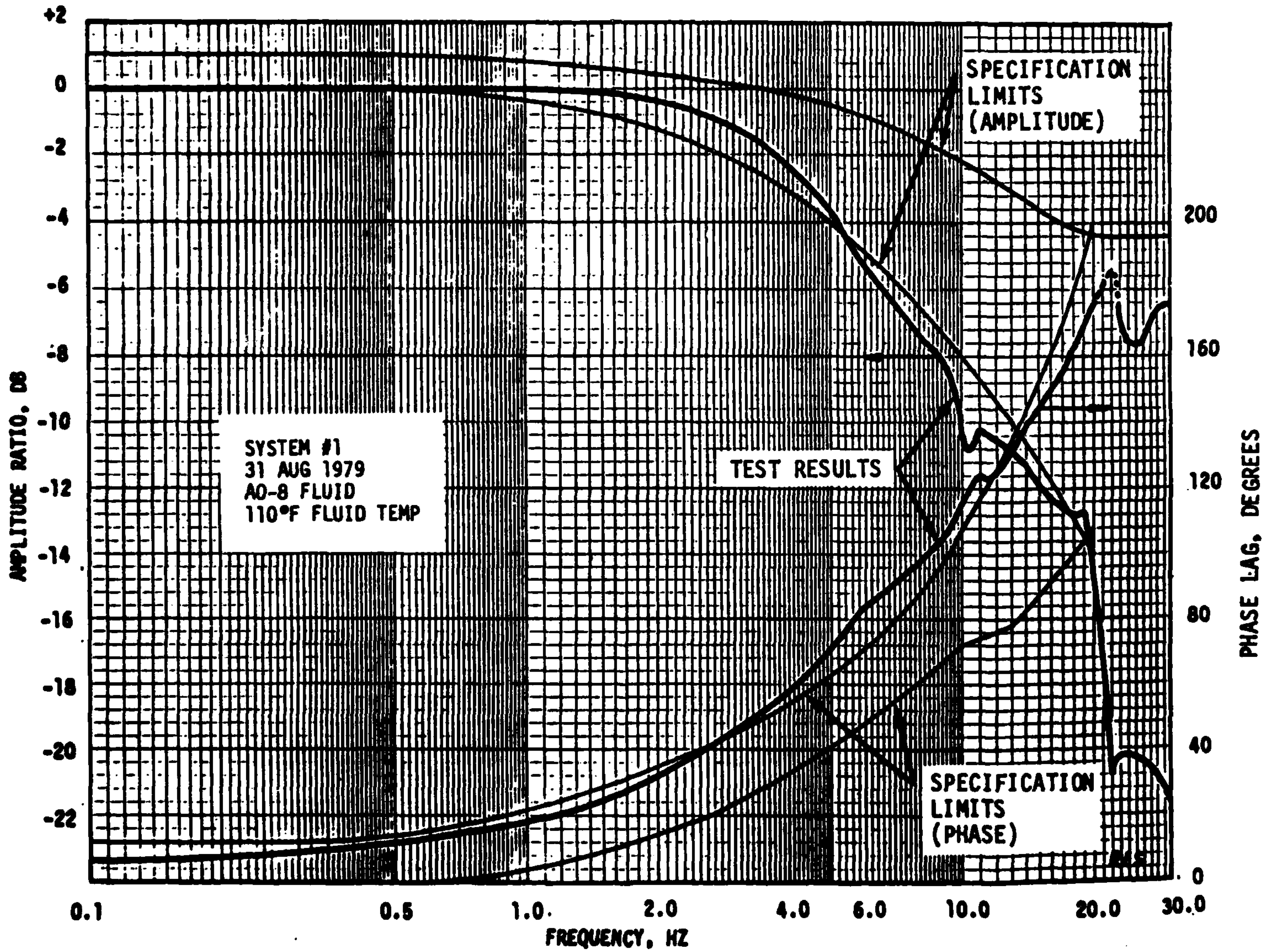


Figure 98. Servoactuator system No.1 dynamic response (pre-endurance)

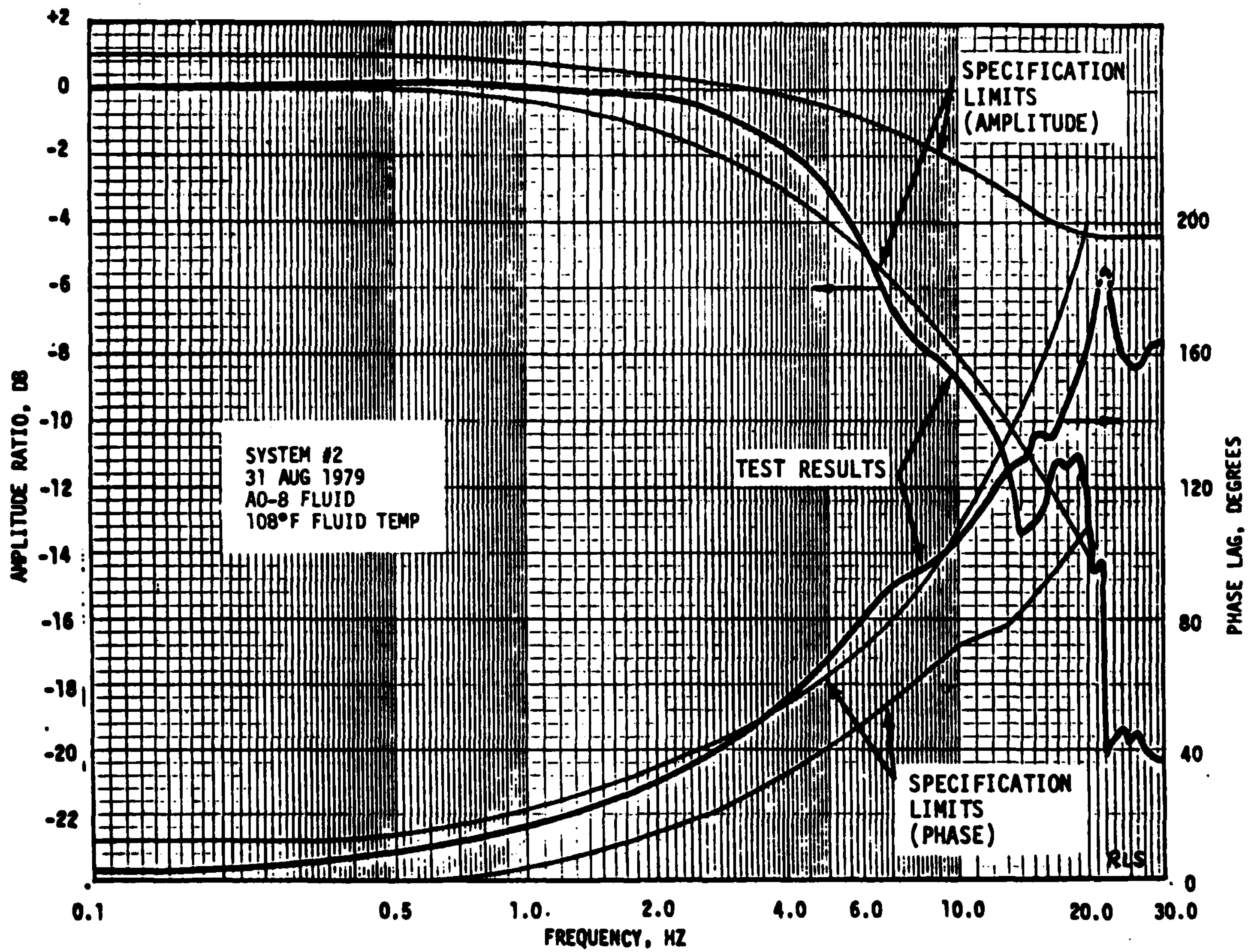


Figure 99. Servoactuator system No.2 dynamic response (pre-endurance)

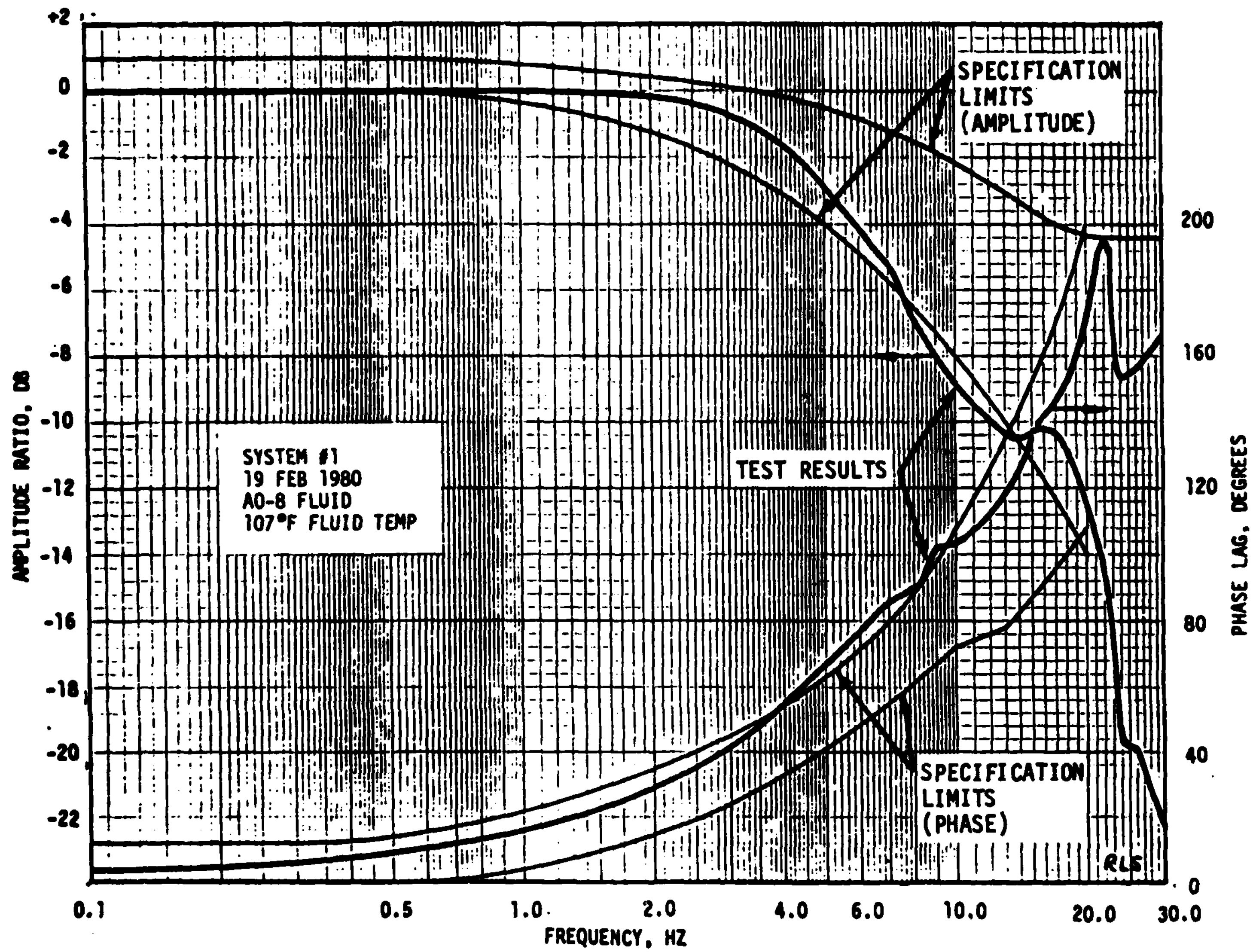


Figure 100. Servoactuator system No.1 dynamic response (post-endurance)

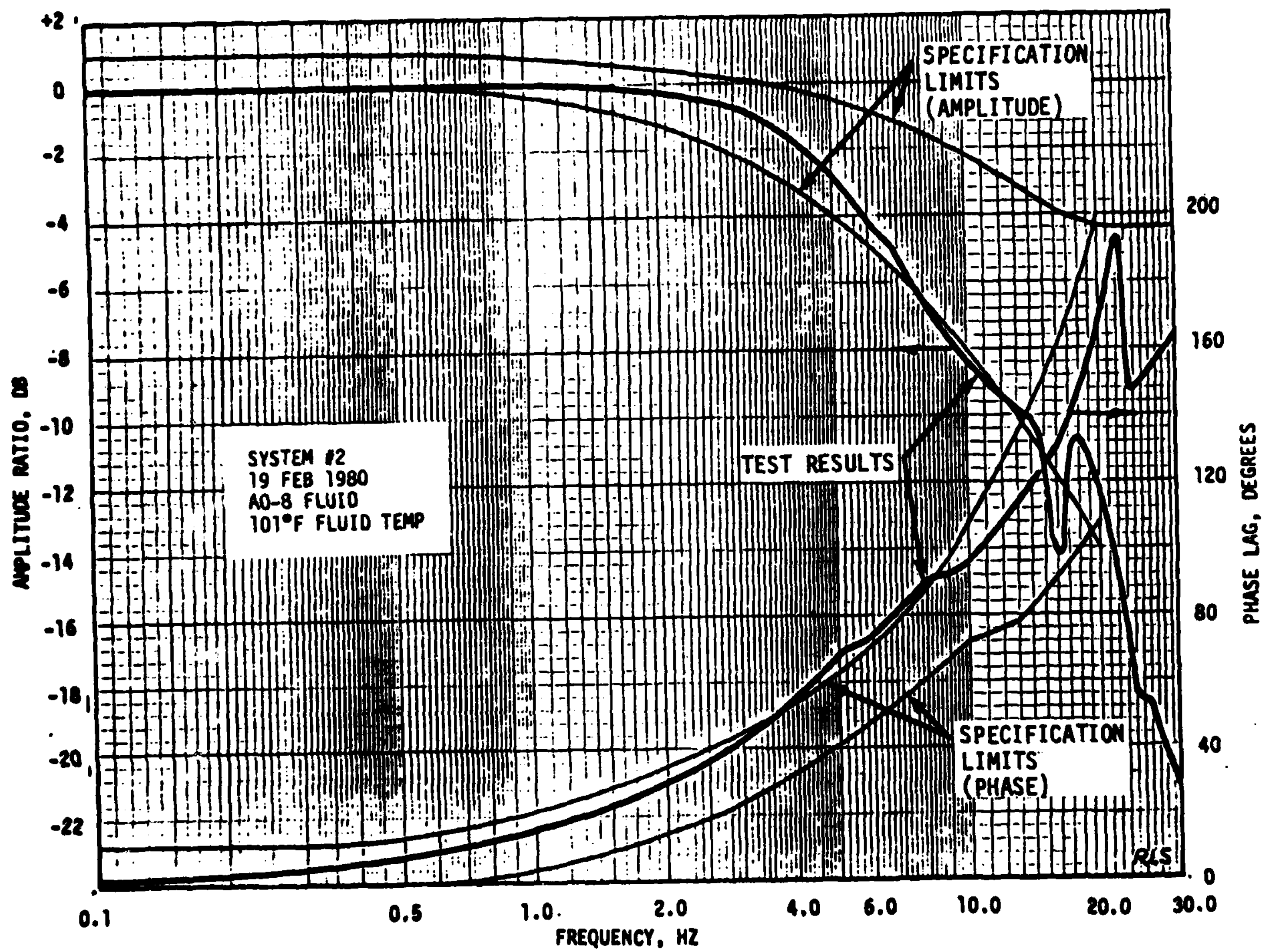


Figure 101. Servoactuator system No.2 dynamic response (post-endurance)

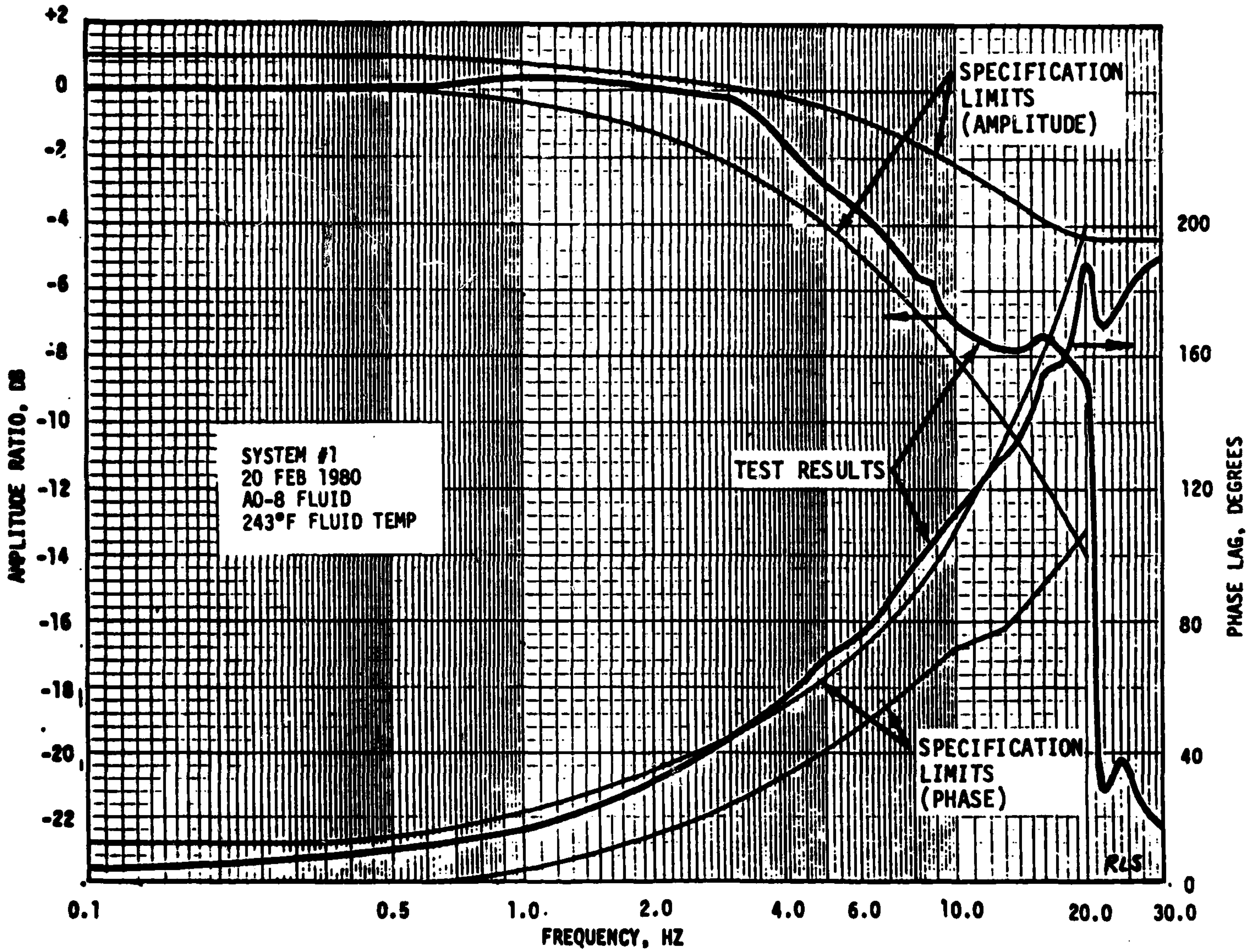


Figure 102. Servoactuator system No.1 high-temperature dynamic response

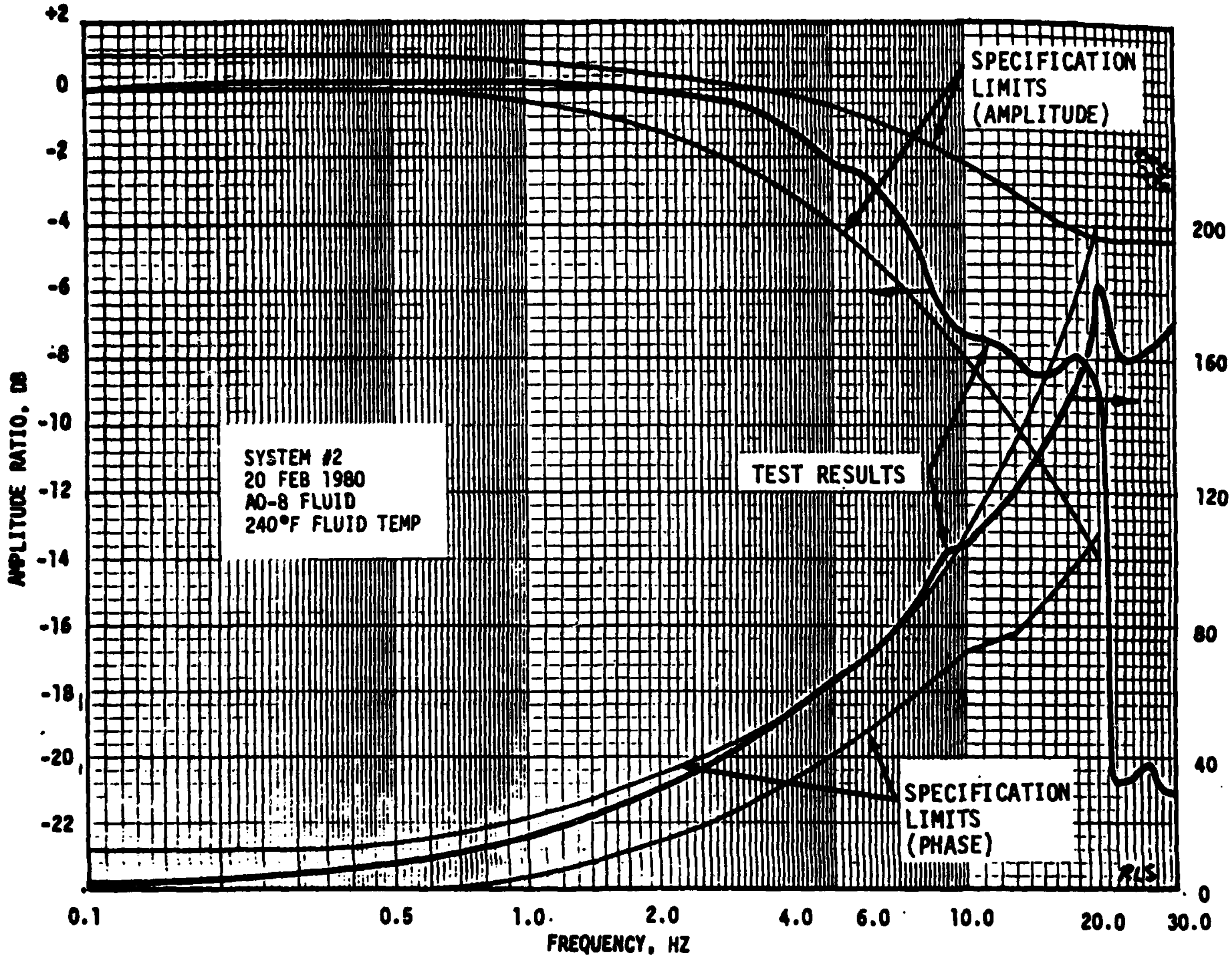


Figure 103. Servoactuator system No.2 high-temperature dynamic response

Servoactuator response at 110F was out of tolerance above 5 to 6 Hertz before endurance testing and above 7.5 Hertz after endurance testing. Response at 240F after endurance testing was within tolerance at all frequencies. Servoactuator phase lag was generally out of tolerance in the 3 to 10 Hertz range throughout testing.

3.3.4.4 Hysteresis Tests

Mechanical and electrical hysteresis tests were performed before and after endurance testing and also at an elevated temperature.

As shown by the following table, the mechanical hysteresis was initially over 6 times the specification limit of 0.008 inch (maximum) and improved after endurance to a value over 3 times the specification. Post-test teardown disclosed no reason for the high mechanical hysteresis.

Test Date	Mechanical Hysteresis (in)	Fluid Temp (F)
4 Sept. 1979	.052	98
19 Feb. 1980	.028	100
20 Feb. 1980	.024	245
Specification Requirement	.008 (max)	100 ±20

Electrical hystersis was initially within the specification limit of 0.020 volts (maximum) and deteriorated only slightly during endurance testing. The elevated temperature hysteresis, however, was approximately twice the specification requirement. Deterioration of the Buna-N servovalve internal packings due to AO-8 fluid exposure is a possible but unconfirmed cause of the increased electrical hysteresis.

Test Date	System	Electrical Hysteresis (volts)	Fluid Temp (F)
30 Aug. 1979	#1	.019	105
	#2	.013	105
19 Feb. 1980	#1	.024	95
	#2	.018	95
21 Feb. 1980	#1	.043	240
	#2	.035	245
Specification Requirement	#1 or #2	.020 (max)	100 ±20

3.3.4.5 Threshold Tests

Mechanical and electrical threshold tests were performed three times during servoactuator testing: once before endurance testing and again after endurance testing, both at ambient and elevated temperatures.

Mechanical threshold was initially well within tolerance but deteriorated to almost four times the allowable value after endurance testing. Cause of the deterioration was not found during the post-test teardown inspection.

Test Date	Mechanical Threshold (in)	Fluid Temp (F)
4 Sept. 1979	.0015	100
19 Feb. 1980	.0155	100
20 Feb. 1980	.0133	246
Specification Requirement	.004 (max)	100 ±20

The electrical threshold was within the specification requirement throughout testing.

Test Date	System	Electrical Threshold (volts)	Fluid Temp (F)
30 Aug. 1979	#1	.0025	98
	#2	.003	102
19 Feb. 1980	#1	.002	100
	#2	.0025	100
21 Feb. 1980	#1	.0015	240
	#2	.0015	239
Specification Requirement	#1 or #2	.0035 (max)	100 ±20

3.3.4.6 Leakage Tests

3.3.4.6.1 Internal Leakage Tests

Internal servoactuator leakage was measured both before and after endurance testing. The internal leakage was slightly increased after endurance testing but was still within tolerance.

Test Date	System	Internal Leakage (cu.in/min) Sol. off / on	Fluid Temp (F)
27 Aug. 1979	#1	14.0 / 33.9	100
	#2	12.8 / 35.1	100
20 Feb. 1980	#1	18.6 / 41.7	65
	#2	18.4 / 40.7	67
Specification Requirement	#1 & #2	20 / 71 (max)	100 ±20

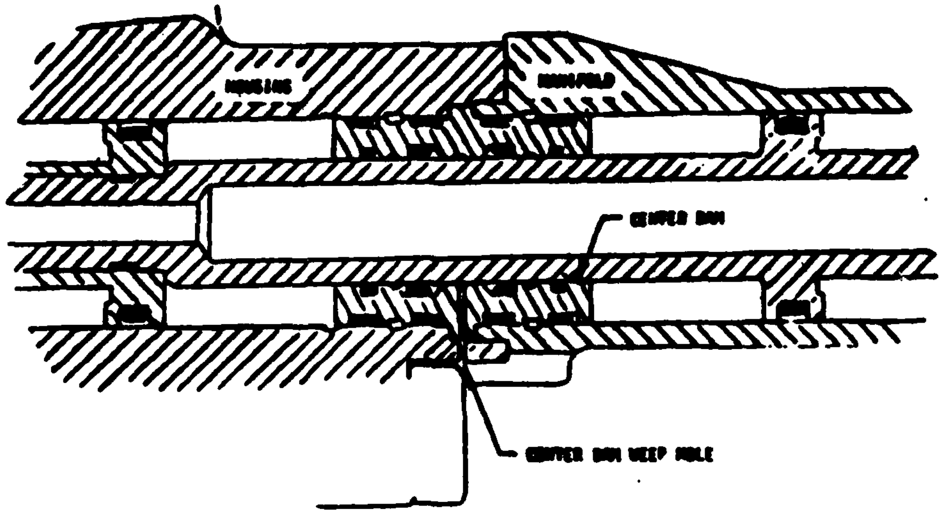
3.3.4.6.2 External Leakage

No external leakage test was performed per se, but external leakage was monitored periodically throughout the endurance testing.

Center dam seal leakage from the weep hole in the housing began early in endurance testing and continued throughout testing. As shown in Table 36, once the center dam leakage began, it averaged ten times the specification limit of 50 cycles per drop.

TABLE 36 SERVOACTUATOR CENTER-DAM LEAKAGE SUMMARY

Test Date	Test Phase	Center-Dam Leakage Full-Stroke Cycles/Drop
	thru 1B	zero leakage
28 Nov. 1979	1C	150
30 Nov. 1979	2B	7
30 Nov. 1979		4.5
3 Dec. 1979		5
18 Dec. 1979		8
2 Jan. 1980		6.5
3 Jan. 1980		5
3 Jan. 1980		5
7 Jan. 1980		4
10 Jan. 1980	3F	4
11 Jan. 1980	4B	6
11 Jan. 1980		5
14 Jan. 1980		8
17 Jan. 1980	4F	4
18 Jan. 1980	5B	5
18 Jan. 1980		4.6
15 Feb. 1980	5F	5
Specification Requirement		50 minimum (1 drop per seal per 100 cycles)



3.3.4.7 Endurance Test

The endurance test was terminated after completion of one-half of the specified endurance cycles due to repeated servoactuator filter-cap failures which caused an impending shortage of AO-8 fluid. Approximately 2,500,000 cycles were completed.

Ten servoactuator filter-cap related failures occurred during the endurance test and are summarized in Table 37. As shown thereon, three different elastomeric O-ring packing materials were used in an attempt to find one that would hold. Those were the phosphonitrilic fluoroelastomer (PNF), hydrofluorocarbon (Viton), and chlorinated polyethylene (CPE). In two of the failures, the filter-cap packing (one a PNF O-ring and the other a CPE O-ring) extruded and allowed fluid leakage without producing filter-cap failures. The other eight failures were due to filter-cap thread failure attributed to excessive swell of the cap packing. Four of those had a PNF O-ring, two had a Viton O-ring, and two had a CPE O-ring.

Figure 104 shows the two failures which occurred on 4 January 1980, but all other caps failed in like manner and location. Calculations indicated that a seal (swell) pressure of 14,000 psi minimum is required in addition to the 3000-psi fluid pressure to fail the filter-cap threads. The calculations were based on a hypothetical filter-cap failure at the 7500-psi servoactuator burst test pressure. Since cap failures did not occur at 7500 psi, the calculated 14,000-psi seal pressure is considered conservative.

There was also an indication that the piston rod foot seal O-ring had also swelled. Visible wisps of AO-8 vapor were seen rising from the very hot piston rod which indicated that O-ring swell had significantly increased the seal friction.

Although the filter-cap failures were attributed to excessive seal swell, there is an anomaly present. Testing by the Materials Laboratory, as reported in Reference 4, showed that, after immersion in AO-8 fluid for 72

TABLE 37. SERVOACTUATOR FILTER-CAP SEAL FAILURES

Failure No.	Date	Hours of Operation	Failure Description
1.	30 Nov. 1979	70:46	No. 2 Servo Filter Cap Packing (PNF) Extruded
2.	18 Dec. 1979	88:26	No. 2 Servo Filter Cap Failure (PNF packing)
3.	4 Jan. 1980	131:15	No. 2 Servo Filter Cap Failure (PNF packing)
4.	4 Jan. 1980	137:00	No. 1 Actuator Filter Cap Failure (PNF packing)
5.	11 Jan. 1980	179:10	No. 1 Actuator Filter Cap Failure (PNF packing)
6.	16 Jan. 1980	219:04	No. 1 Actuator Filter Cap Failure (Viton packing)
7.	18 Jan. 1980	254:01	No. 1 Actuator Filter Cap Failure (Viton packing)
---	12 Feb. 1980	254:01	Changed all Filter Cap Packings to CPE O-rings
8.	12 Feb. 1980	260:45	No. 2 Actuator Filter Cap Failure (CPE Packing)
9.	14 Feb. 1980	272:09	No. 2 Servo Filter Cap Packing (CPE) Extruded
10.	20 Feb. 1980	282:30	No. 2 Servo Filter Cap Failure (CPE Packing)

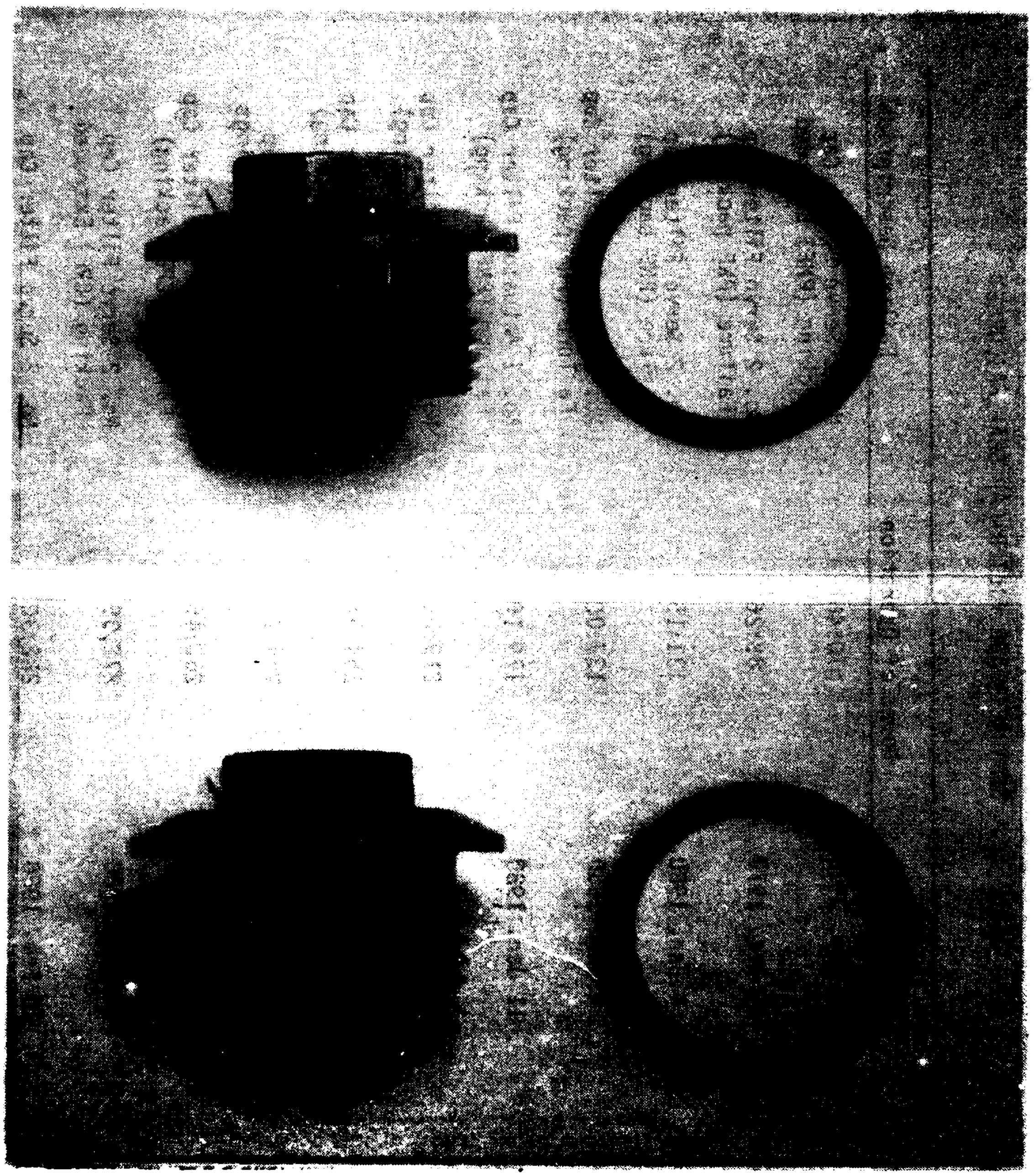


Figure 104. Typical servoactuator filter cap failures

hours at 275F, the PNF and Viton elastomers both swelled approximately 35%. That much swell could have created sufficient pressure to crack the filter-cap threads. However, during aging under the same conditions, the CPE material swelled only 7.9%: an amount normally considered acceptable.

Fluid additions to the servoactuator test stand are summarized in Table 38. Most fluid additions were required because of the numerous filter cap failures.

The temperature history of the servoactuator during endurance testing is as follows:

Temperature (F)	Test Time (Hours)
100 - 130	247
131 - 150	36

The fluid temperature exceeded the test plan value of 100 $\pm$20F because the system heat input during mechanical cycling exceeded the heat removal capacity of the test stand heat exchanger. The fluid temperature during the electrical cycling stabilized in the 115 to 120F range.

3.3.4.8 High-Temperature Test

High-temperature testing was performed at a nominal fluid temperature of 240F and at normal ambient air temperatures. Dynamic response, hysteresis, and threshold tests were performed. The results of those tests have been previously discussed under the specific test sections. Leakage tests were waived at the 240F fluid temperature because of the hazards of collecting fluid samples at that temperature.

3.3.4.9 Low-Temperature Test

A low-temperature test was performed at a nominal fluid temperature and actuator ambient temperature of -65F. Directional response to an electrical input command was the only function checked since here is no specification requirement for operation at fluid temperatures below -40F.

Actuator response to a 7.78-volt peak-to-peak 1-Hertz square wave at a temperature of -61F is shown in Figure 105. Figure 106 shows the response to the same input command at a fluid temperature of 77F for reference.

The delay of approximately 0.75 seconds from the initial command input at -61F to the actuator response was the delay between the command input and the solenoid valve being energized. Actuator response was sluggish and overshot the command for the first cycle and a half, but rapidly improved thereafter.

3.3.4.10 Post-Test Teardown Inspection

The servoactuator was totally disassembled on 29 April 1980 for inspection after termination of endurance testing at 2,520,000 cycles.

TABLE 38. TEST STAND FLUID ADDITIONS

	FLUID ADDITION (QTS)	
Date	Servoactuator Stand	Long-Term Pump Stand
14 Aug. 1979	----	16 (1)
20 Aug. 1979	8 (1)	----
28 Aug. 1979	1	----
24 Oct. 1979	----	3
12 Nov. 1979	----	1
30 Nov. 1979	2	----
18 Dec. 1979	4	----
2 Jan. 1980	----	2
4 Jan. 1980	5	----
4 Jan. 1980	2	----
11 Jan. 1980	3 1/2	----
16 Jan. 1980	3	----
11 Feb. 1980	5 1/2	----
12 Feb. 1980	4	----
14 Feb. 1980	2 1/2	----
20 Feb. 1980	1	----
20 Feb. 1980	1	----
27 Feb. 1980	4	----
21 Mar. 1980	----	3 1/2 (2)

(1) Initial Fill

(2) Initial Fill - Cold-Start Stand

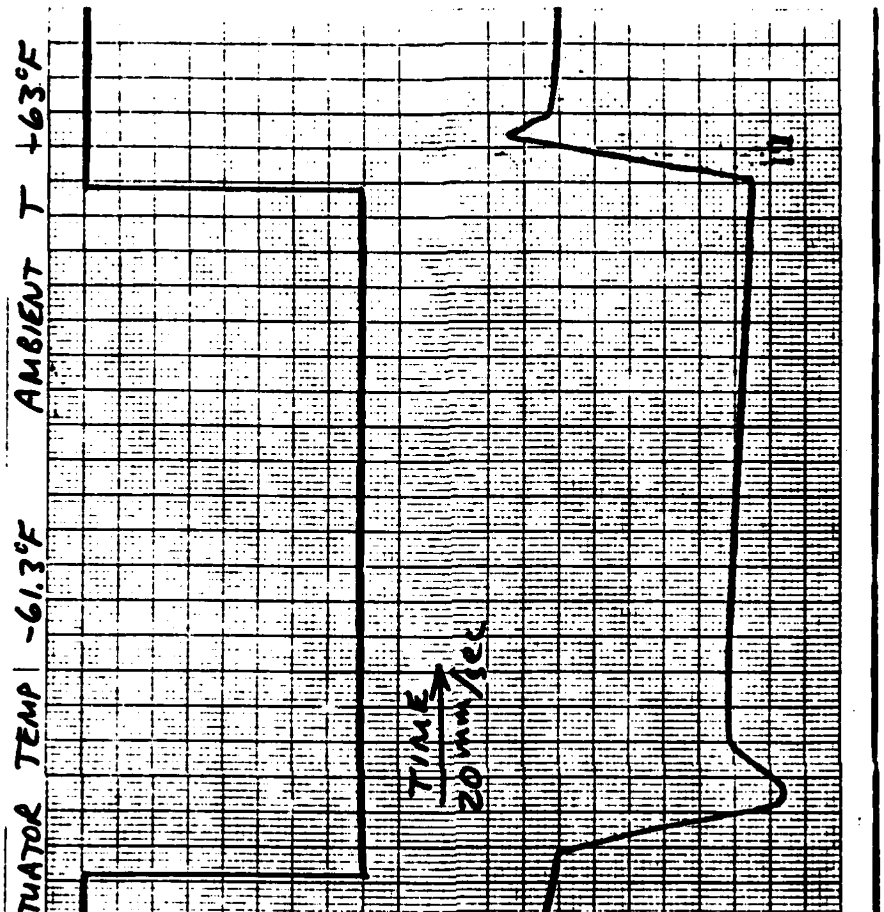


Figure 105. Servoactuator response at -65F

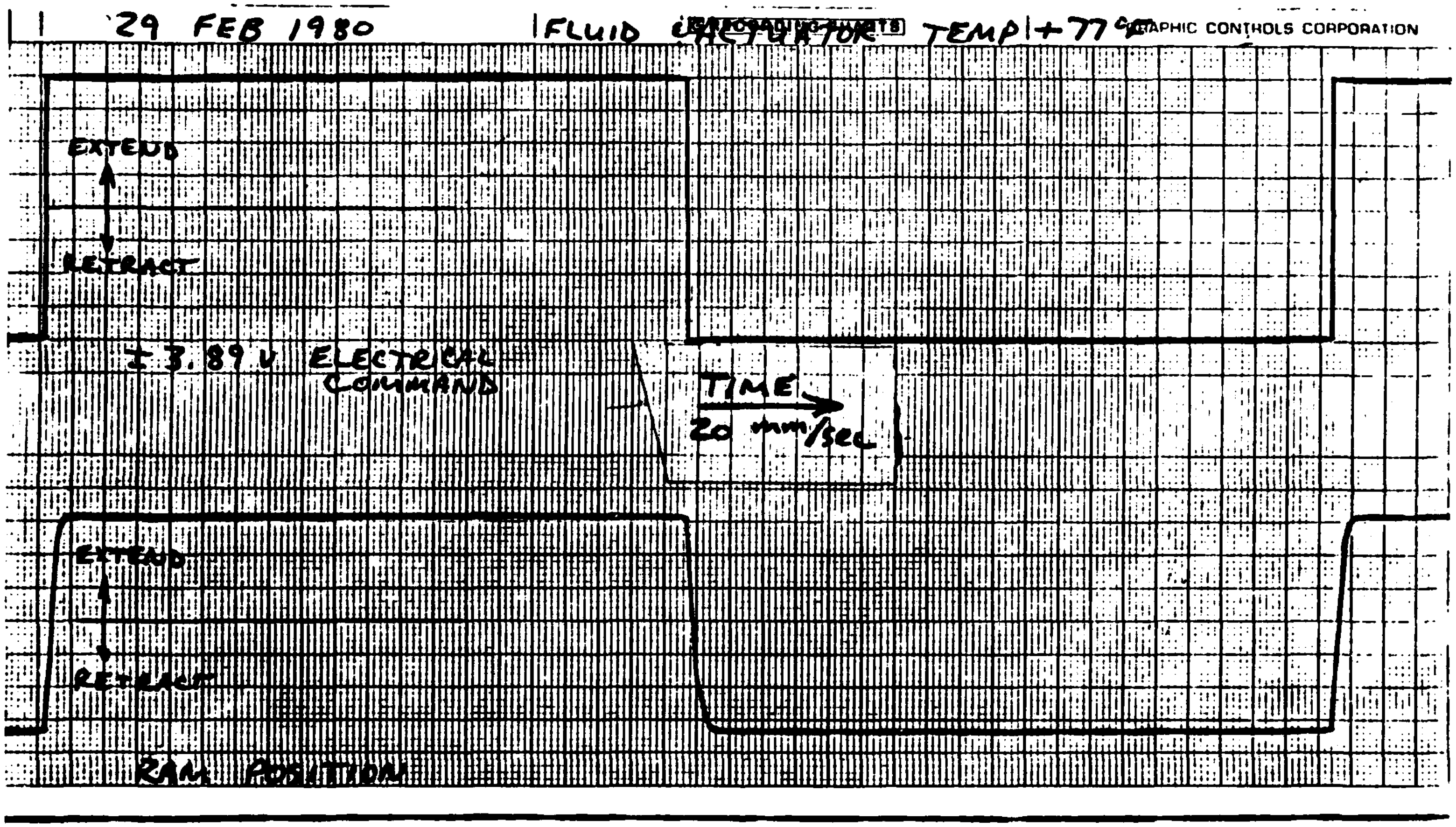


Figure 106. Servoactuator response at 77F

3.3.4.10.1 Seal Condition

All PNF seals were slightly softer than when originally installed, but were not as soft as those observed throughout the pump test programs. The limited operation of the servoactuator at high temperatures (2 hours @ 250F) implies that the softening of the PNF material is temperature dependent.

The System No. 1 servovalve filter-cap packing was found partially extruded during teardown. No leakage had yet occurred.

The control valve sleeve packings were soft and slightly nibbled on the outer diameters. The damage was probably due to removal from the actuator housing. Figure 107 shows seal condition after removal.

The System No. 2 piston seal and packing were extensively damaged. The sealing web of the W. S. Shamban Co. Double Delta Channel Seal, made of their Turcon (filled-Teflon) material, was completely worn through and was approximately 70 percent missing. The PNF O-ring, which provides radial support to the channel seal, was badly nibbled, separated into two pieces, and approximately 50 percent missing. Figures 108 and 109 depict the seal condition at teardown.

The failure of this piston seal was attributed to wearout damage of the Turcon ring, and then the O-ring, due to rubbing against the roughened cylinder bore of this previously used actuator unit (described in 3.3.4.1). The relatively high swell of the PNF material could have increased the radial force on the seal against the cylinder wall thereby increasing the wearout rate. In contrast, the companion seal on the System No. 1 piston was found in excellent condition.

The teflon multi-turn spiral backup rings in the manifold and center dam were partially extruded, one slightly and two severely. The damage has been attributed to improper installation by the Sacramento ALC technician.

The System No. 2 servovalve coil housing was full of AO-8 fluid at disassembly. The Buna-N seals were intact but hardened, thus allowing leakage into the coil housing.

3.3.4.10.2 Steel Discoloration

The control valve sleeve and shuttle were the only steel components which exhibited any significant discoloration (Ref. Figure 107). The steel sleeve was discolored over its external surface except under the packings, and the shuttle was discolored over all surfaces except the lands. Figure 110 shows the control valve shuttle before and after exposure to the AO-8 fluid.

One auxiliary actuator piston exhibited slight discoloration which could be removed by rubbing with a finger. All other steel components such as the linkage and by pass valves were also slightly discolored.

3.3.4.10.3 Piston Rod Wear

The piston rod diameter was not measured before or after test. Insignificant wear occurred, however, since only one small area on the rod exhibited a slight polish. The remainder of the rod exhibited no contact whatsoever.

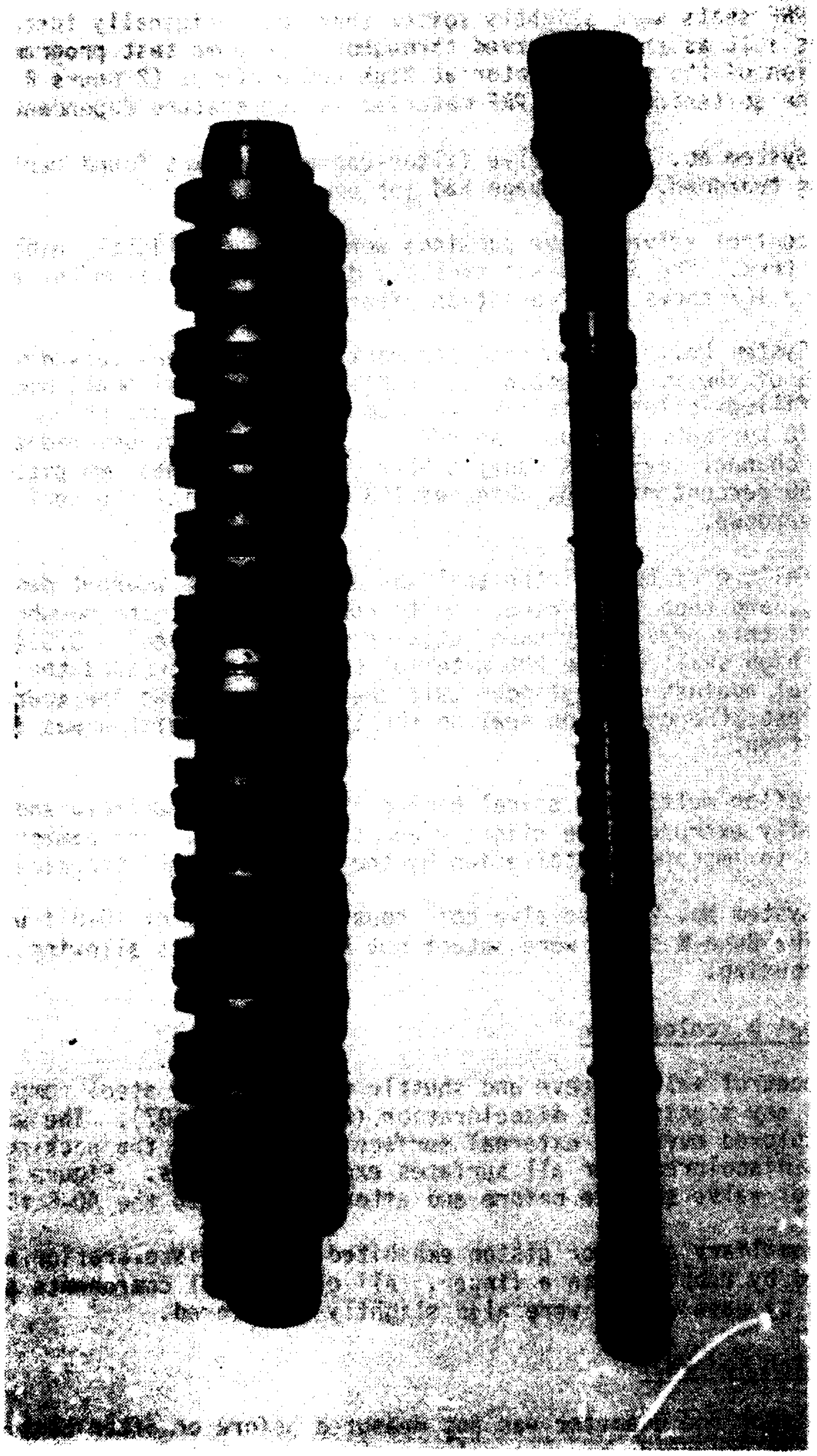


Figure 107. Control valve spool and sleeve

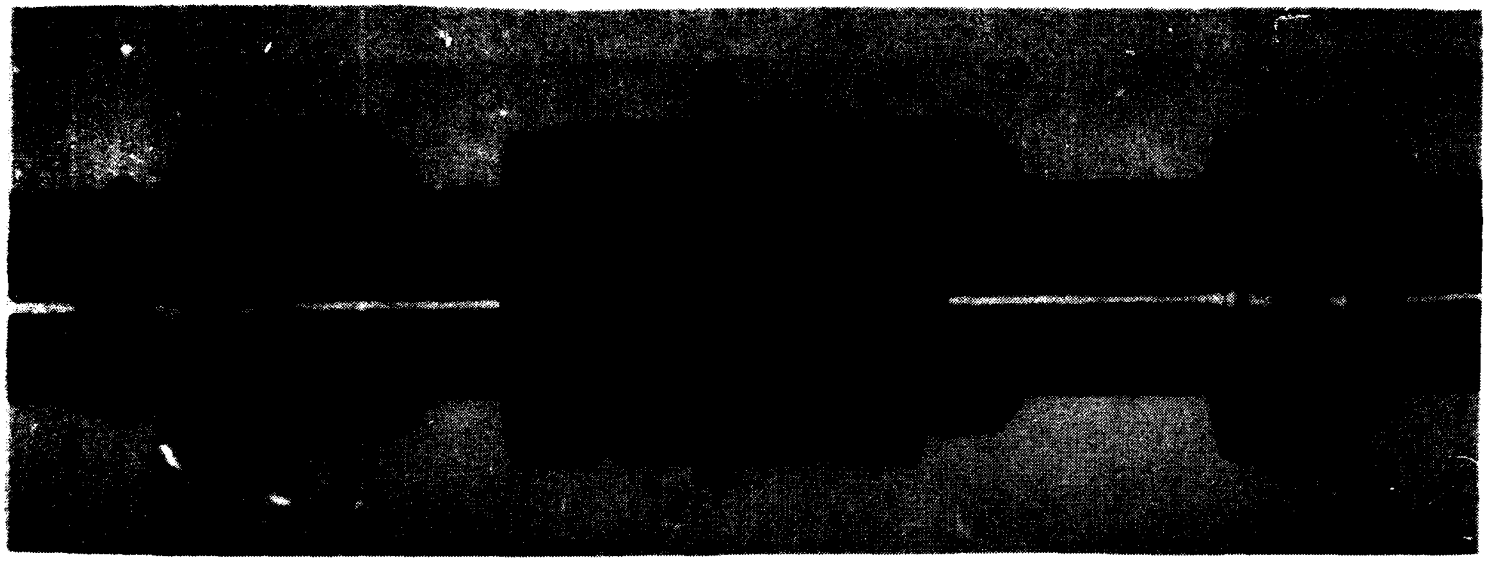


Figure 108. Piston and center-dam seals (No.2 on left)

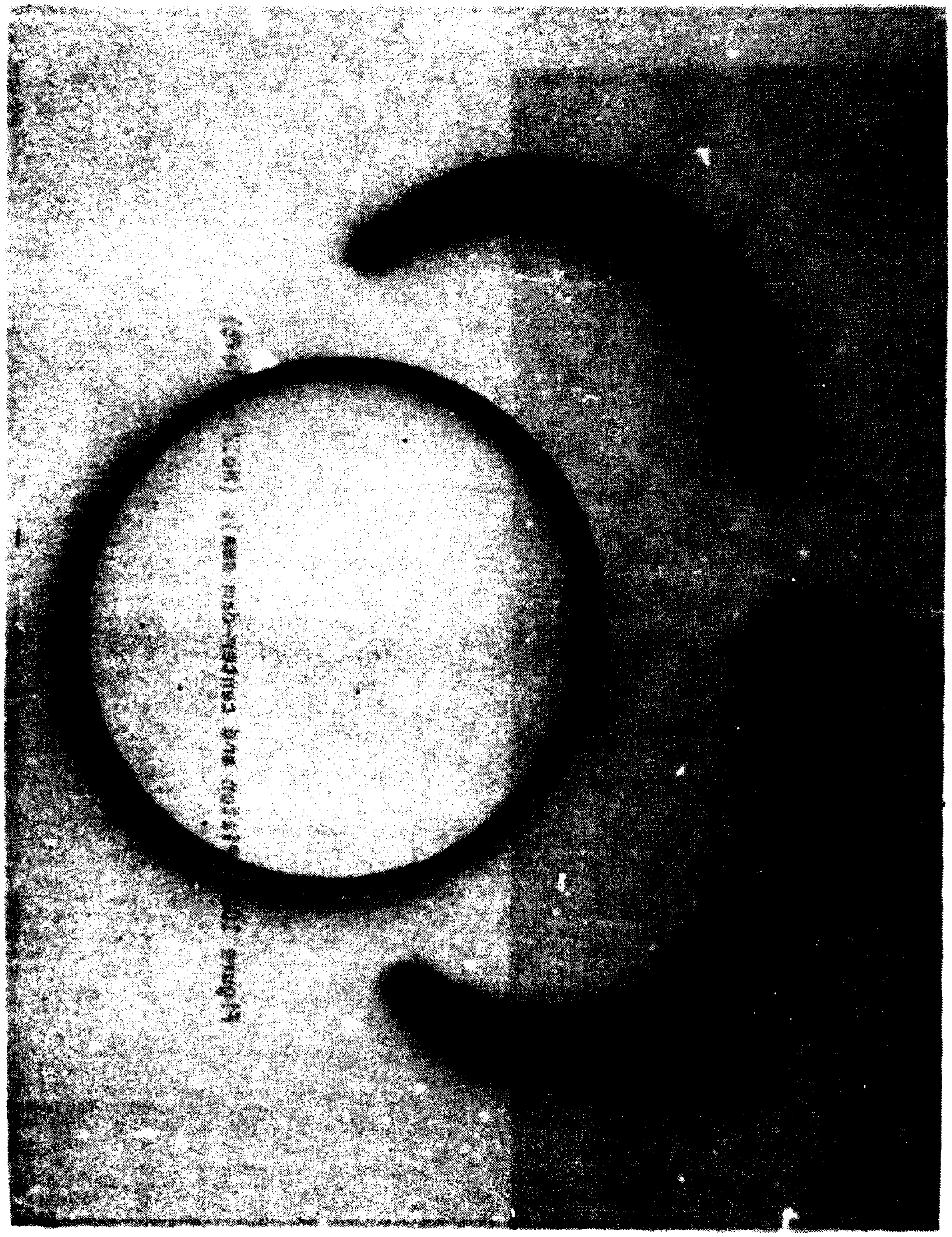


Figure 109. System No.2 piston seals

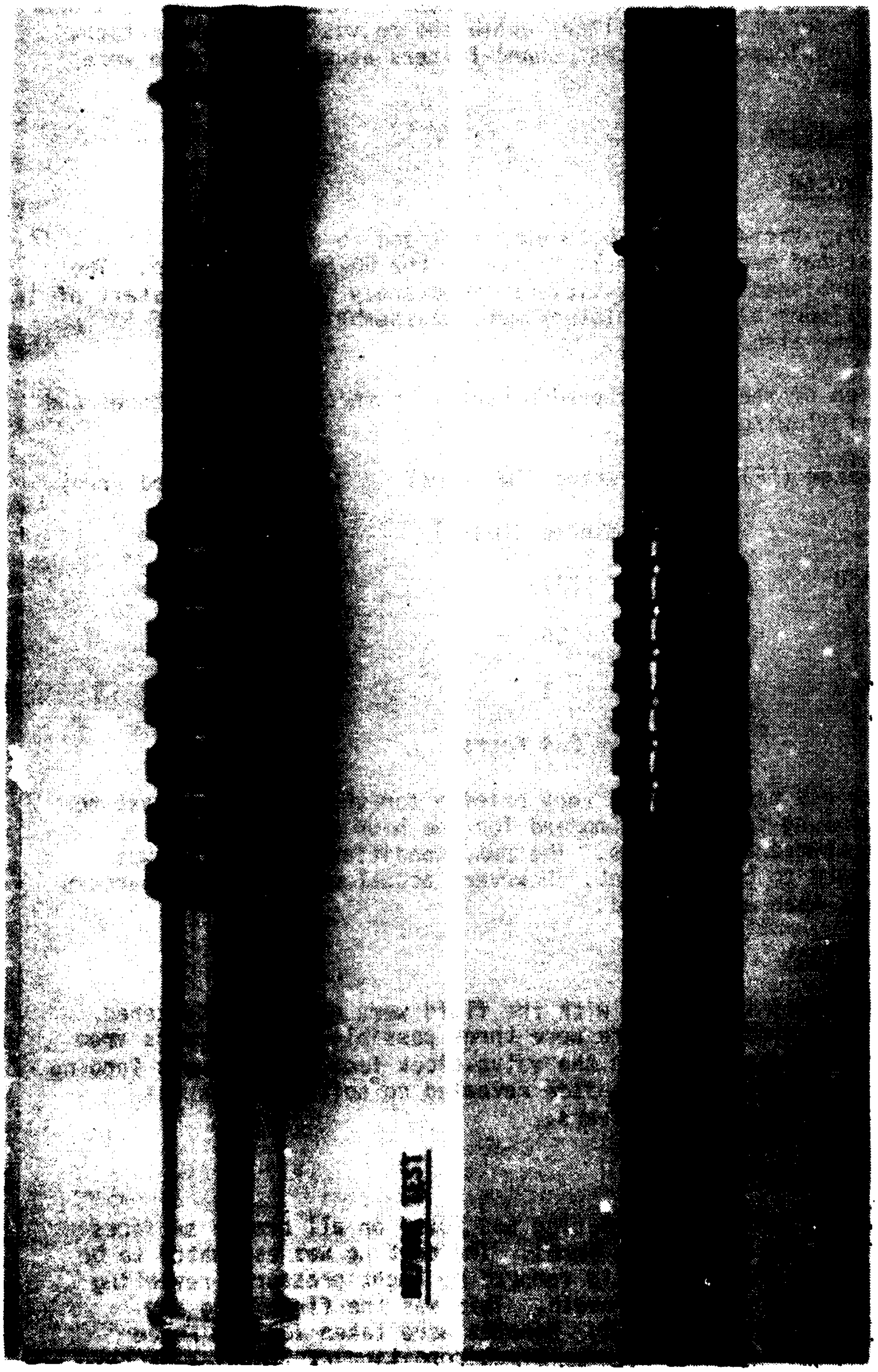


Figure 110. Control valve shuttle before and after test

3.3.4.10.4 Filter Condition

The four servoactuator filters exhibited no visible contamination. The rubber particles found in the test stand filters at stand teardown were actuator generated.

3.3.4.11 Pump Condition

3.3.4.11.1 Background

The Sperry-Vickers PV3-075-19 pump utilized to operate the servoactuator test had been previously tested as the 50-hour test pump. The pump had been refurbished by Sperry-Vickers immediately prior to the start of this test (new cylinder block and piston/shoe subassemblies). New PNF elastomer seals were also installed.

Operation of the pump was predominantly at moderate temperatures and speeds. See the following table.

Temperature Range (°F)	Operating Time (Hrs)	Operating Speed (rpm)
-65	10 minutes (max.)	0 - 7000
100 - 130	247	2500
131 - 150	36	2500
240 - 260	1	2500
	Total = 284 Hours	

The pump had been utilized very briefly for the cold-start test to replace the failed long-term test pump and for one hour to perform the servoactuator high-temperature tests. The pump condition at teardown was therefore anticipated to be excellent. However, actual condition at teardown was quite different than anticipated.

3.3.4.11.2 Valve Block

Steel surfaces in contact with the fluid were generally darkened similar to previous teardowns. There were three possible bronze smears from the cylinder-block Kingsbury pads on the valve-block face. Subsequent lapping and nital etching of the valve-block face revealed no soft spots as was initially indicated by the bronze smears.

3.3.4.11.3 Cylinder Block

A heavy dark brown/black coating was found on all bronze surfaces except the valve face and the piston bores. The coating was estimated to be .010 to .015 thick and could be easily removed by light pressure, revealing the more typical discoloration underneath. This was the first time any coating of this nature had been found. Samples were taken for Air Force analysis. Figure 111 shows the cylinder-block condition at Sperry-Vickers before and after testing.

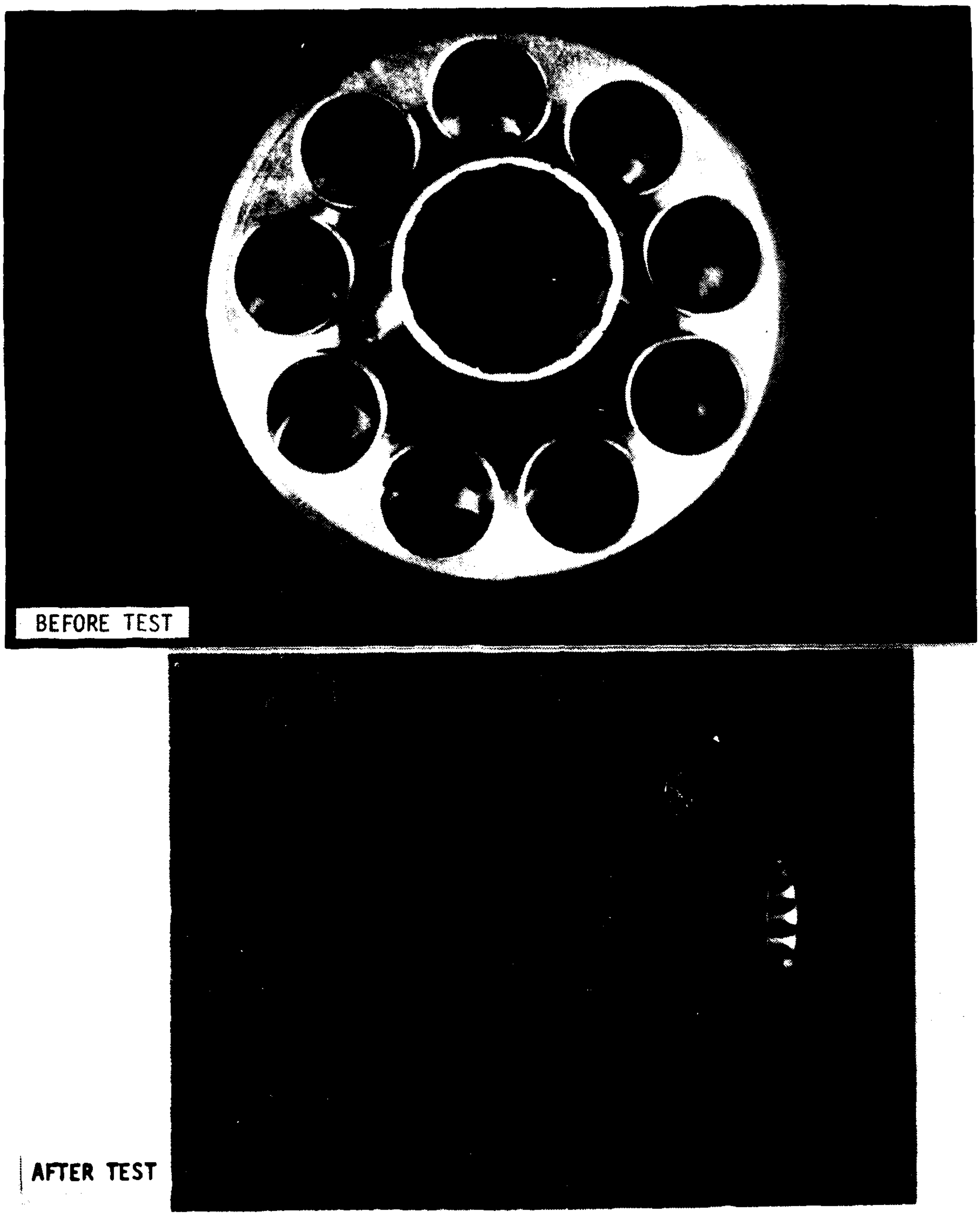


Figure 111. Pump cylinder block before and after testing

Cylinder-block piston bores were measured and found to be out-of-round and tapered. A comparison of the bore diameters indicated that a coating had also been formed on the cylinder-block bores.

Average bore dimensions after test were:

	Top	Middle	Bottom
Average Bore Dia	0.44895	0.44933	0.44923
Average Out-Of-Round	0.00012	0.00024	0.00026

How completely the bore coating was removed prior to measurements could not be determined.

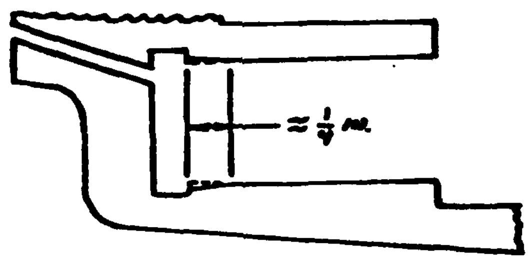
3.3.4.11.4 Actuator Piston

A slight polishing of the piston end which enters the housing bore was noted. There was no evidence of scratches or galling.

The piston was very lightly discolored over all surfaces.

3.3.4.11.5 Housing

The actuator piston bore appeared "belled" out at the bottom end - see sketch. The defect was apprently not detrimental and was probably due to manufacturing since a worn hard-coat would also have damaged the piston.



The condition of the anodized surfaces was typical with no discoloration or degradation noted.

3.3.4.11.6 Piston/Shoe Subassemblies

All non-wearing surfaces of the bronze shoes were coated with the same thick brown/black coating noted previously on the cylinder block.

No erosion damage to the shoe faces was evident. Shoes #5 and #8 appeared out-of-flat or warped. Shoe #6 had a shiny "worn" spot on one side. The remaining shoes had radial "chatter" marks on the faces.

The pistons were slightly discolored and several had a polished band approximately 0.9 inches from the end.

A comparison of the piston-bore charts indicated that the pistons had grown in diameter by 0.0001 to 0.0002 and had become slightly out-of-round.

3.3.4.11.7 Thrust Bearing

The bronze ball spacer had a brown/black coating on non-wearing surfaces.

The beaing outer race had a "heat" ring around the entire circumference approximately in line with the ball load path. The brownish-blue color requires exposure to a temperature of 400 to 600F to be produced in air. The temperature required to produce the same color in AO-8 fluid is unknown.

The ball paths on the outer race ID and the inner race OD were also discolored but appeared more like corrosion.

3.3.4.11.8 Mounting Flange

The thrust-bearing bore had an insoluble brown varnish-like stain approximately one-inch long in line with the discoloration of the bearing outer race. No other anomolies were noted.

3.3.4.11.9 Miscellaneous Bronze Parts

The shoe retainer-plate surfaces were coated with the dark scale. The non-contact bronze surfaces and chrome plated areas of the shaft seal not in contact with the PNF grommet were also coated with the scale. The seal wearing surface was discolored.

3.3.4.11.10 Sperry-Vickers Teardown Inspection Report

The Sperry-Vickers teardown inspection report is included as Appendix K.

3.3.4.12 Fluid Condition

3.3.4.12.1 Fluid Analyses

All filters utilized in the servoactuator test stand were standard 5-micron absolute rated units provided by Aircraft Porous Media. The standard housing seals had been replaced with PNF seals, but the filter elements were fabricated with their standard materials and epoxy. See Appendix L.

Fluid samples were taken upstream of the pump discharge filter during endurance test. Particle counts and chemical analyses were performed on the samples by the Scientific Laboratory Service of the Pall Corporation and by the Materials Laboratory at Wright-Patterson AFB, respectively.

Particle count results are presented as follows even though the results are considered meaningless because of the frequent fluid additions to the test stand.

Date	Servoactuator Hours	Discharge Effluent Cleanliness, NAS-1638 Class
16 Nov. 1979	5	6
7 Dec. 1979	70.6	6
10 Jan. 1980	175.6	3

The pump return filter bowl at the end of the test contained a large quantity of rubber and teflon particles, obviously generated by failure of the actuator Number 2 piston seal and the piston-rod foot seal.

3.3.4.12.2 Fluid Color

Color of the new AO-8 fluid was clear, almost water-white, but with a slight yellow tint. Color of the test-stand fluid in October 1979, after the stand sat idle for one and a half months, was a very definite yellow. Test-stand fluid color at the end of the servoactuator test was again water-white, probably due to the frequent fluid additions.

3.3.5 Servoactuator Test Conclusions

The modified servoactuator generally fulfilled all performance expectations, including acceptable dynamic response comparisons with previous test data. The modified control-valve metering slots as defined in the analog definition phase successfully maintained the required servoactuator flow gain with the denser AO-8 fluid.

Discoloration of steel surfaces by the AO-8 fluid was not apparently detrimental to the servoactuator in the short term. However, additional testing would be required to evaluate long-term effects.

The two filter-cap seal extrusion failures, and the eight structural failures of those caps which led to the premature termination of endurance testing, were the primary problems observed. As noted in 3.3.4.7, those failures were attributed to excessive swelling of the O-ring packing under the filter cap. Previous testing by the Materials Laboratory showed that two of the three O-ring materials used in this application (PNF and Viton) did have a high swell characteristic (approximately 35%) in AO-8 fluid. However, the three failures with the chlorinated polyethylene (CPE) O-rings would not be anticipated by the swell value (7.9%) measured during the Materials Laboratory tests. It should be noted that this has not been a problem with these servoactuators in the B-52 fleet using MIL-H-5606 fluid and MS28778 O-rings.

4.0 OVERALL CONCLUSIONS

The stringent nonflammability goals developed for future hydraulic fluids by the Aero Propulsion Laboratory and the Aeronautical Systems Division severely limited the types of materials which could be considered. Among the chemically different types of candidate nonflammable fluids, the organic compounds with high fluorine content (ethers and chlorofluorocarbons) appeared to have the best combination of desirable fire performance capabilities, ie: a high ignition temperature, low propensity to propagate flames, and a low heat of combustion.

The comprehensive survey conducted to identify materials for further evaluation revealed only two candidates worthy of serious consideration. Others were proposed which appeared to meet the nonflammability goals; but, they either lacked one or more properties required for a practical hydraulic fluid, or their projected production price was too high.

The Halocarbon Products Corporation's AO-8 chlorotrifluoroethylene (CTFE) fluid appeared to be a better choice than the other candidate: DuPont "Freon" Products Division's "Freon" E6.5 fluorinated ether. Except for differences in viscosity, bulk modulus, and elastomer compatibility, the properties of the two fluids were essentially equal; and, they are both relatively heavy and expensive. However, the projected price and investment cost to produce the E6.5 fluid was considered unacceptable; and, therefore, the AO-8 fluid was recommended for component compatibility testing.

Analyses and testing conducted to date indicate that the AO-8 fluid has potential for use as an aircraft hydraulic fluid, that satisfactory component performance can be obtained, and that its use should completely preclude hydraulic fluid fires. However, systems and components must be designed to accommodate its high density and viscosity. For a conventional 3,000 psi system, this requires the enlargement of most component orifices and valve slots, the derating of hydraulic pump speeds, increased reservoir pressures, and the use of larger line sizes. These factors, and the increased weight of the fluid itself, add up to a significant increase in system weight if the fluid is used throughout a system of the conventional type.

One possible way to reduce the weight penalty is to use a lower-viscosity CTFE fluid which would allow reductions in tubing size and fluid volume. Another approach is to use the CTFE fluid only in those subsystems which are proximate to ignition sources such as engines and wheel brakes. Tests already completed on a two-fluid brake system indicate the concept is feasible and that it would entail only a moderate increase in weight. Other means, such as the use of higher pressures or load-adaptive actuation systems to reduce fluid flow and thereby reduce component and tubing sizes, can also be considered.

The high cost of the fluid, the increase in costs for component development, the increase in investment costs for new hydraulic benches and ground carts, and the increased use of aircraft fuel due to the increase in system weight, all add up to a significant increase in aircraft life-cycle cost if the fluid is used throughout a system of the conventional type.

The phosphonitrilic fluroelastomer was selected as the O-ring seal material for the component compatibility tests. This material was recommended as the best candidate by AFWAL/MLBT. Component tests indicate a need for additional evaluation and/or development of elastomers for hydraulic seals which can provide acceptable performance and life in CTFE fluid.

Additional evaluation of the fluid's reaction with bronze and other copper-bearing alloys is also required. The discoloration and buildup of surface film is evidence of possible chemical attack; however, the long-term effects are as yet unknown.

In summary, it is concluded that the Halocarbon AO-8 fluid is the best potential hydraulic fluid material available to meet the nonflammability requirements, but that additional development and evaluation is required before it can be recommended for aircraft use. It is also concluded that its use could entail significant support-cost investment and increase aircraft life-cycle cost depending upon how extensively it is used. Its use in complete aircraft systems of the conventional types will add considerable weight. However, several promising possibilities for reducing the weight penalty have been identified.

5.0 RECOMMENDATIONS

The following recommendations for additional work aimed at developing affordable means to utilize the nonflammable property of the CTFE hydraulic fluid in order to reduce the incidence of aircraft hydraulic fluid fires are offered.

5.1 Nonflammable Hydraulic Fluid Development

It is recommended that the evaluation and development of a nonflammable CTFE fluid be continued.

5.2 Elastomer Seal Evaluation and Development

It is recommended that evaluation and development of elastomers for use as hydraulic seals in CTFE hydraulic fluid be continued.

5.3 Establishment of One or More Additional Fluid Source(s)

It is recommended that one or more additional fluid sources be established in order to reduce cost. At the time that the Halocarbon AO-8 fluid was selected over the DuPont E6.5 fluid, in 1977, the Halocarbon Products Corporation's projected cost for quantity production of one million gallons of AO-8 fluid per year in the 1980-1985 time period was $55 to $75 per gallon which was a reduction from the $120 per gallon price estimated in 1976. The increased operation and support costs for a typical twin-engine transport noted in Tables 16 and 17 were based on a price of $75 per gallon. However, since that time, Halocarbon has increased their predicted production price to $125 per gallon.

5.4 Exploration and Analysis of Means to Reduce Weight Penalties

It is recommended that means to reduce the relatively large weight penalty which would be incurred with the use of a CTFE nonflammable fluid in a complete hydraulic system be further explored. Analyses should be made to determine the relative weight-saving potential of the various methods considered.

Several weight-reduction methods are already identified in Subsection 2.3.6 herein; and, the weight savings which might be realized with one of the low-viscosity CTFE fluids in a transport aircraft is summarized in Table B1 in Appendix B.

Other means suggested for reducing weight are use of higher pressure, load-adaptive actuation techniques, satellite systems and/or integrated actuator packages. These approaches offer weight reduction potential by providing smaller components/systems or by reducing tubing lengths. The weight savings associated with these approaches could provide the incentive to obtain the fire safety offered by the CTFE fluid in place of the conventional system for new aircraft.

5.5 Validation of Weight Minimization Concepts

It is recommended that the most promising schemes be validated by

tests as necessary to determine their acceptability. For instance, before a low-viscosity version of the CTFE fluid can be accepted, it must be proven that acceptable component performance and life can be obtained. Obtaining adequate life with hydraulic pumps and motors operating at their maximum fluid inlet temperatures, where fluid viscosity and hydrodynamic lubrication film thickness is lowest, may be difficult. Some pump suppliers have predicted that, even with harder materials such as tungsten carbide, lower operating life will be obtained. Reducing the rated speeds may help; but, considerable development and testing appears necessary before the overall impact of new materials and deratings on weight and cost can be determined.

The use of higher operating pressures may be favored as a means to reduce the weight penalty. However, the use of a low-viscosity fluid in higher pressure systems will probably aggravate the pump and motor problems. Thorough testing, and realistic assessments of the impact on component life, cost, and system performance will be required before that combination can be considered for future aircraft designs.

5.6 Fireproof Brake Hydraulic System Development

It is recommended that development of the two-fluid wheel brake hydraulic system concept be continued. That concept could be used in the very near future to immunize both existing and future aircraft from the most common cause of hydraulic fires: fluid leaking on hot brake stacks. The feasibility of the concept was demonstrated in the Fireproof Brake Hydraulic System program. As noted in the final report, Reference 6, it was recommended that continued development should include the following design and testing efforts (for specific aircraft):

a. Perform a design study to develop the optimum two-fluid brake system design considering all performance factors identified in (that) report. This would include determining required system modifications to achieve stopping performance comparable to the baseline aircraft.

b. Perform laboratory tests (using brake system hardware) to tune the antiskid controller and verify brake system stopping performance. In addition, laboratory tests should be performed to demonstrate the replenishing system and servicing procedures.

c. Perform aircraft ground roll tests to verify and demonstrate the two-fluid brake system performance exposed to a variety of field conditions.

It should be noted that the referenced program indicated that a two-fluid brake hydraulic system configured for the KC-135 aircraft (with appropriate modificaitons to hydraulic lines, etc., to maintain the current stopping performance) would increase the airplane weight by only 64 pounds.

6.0 REFERENCES

1. E. T. Raymond, Design Guide for Aircraft Hydraulic Systems and Components for Use With Chlorotrifluoroethylene Nonflammable Hydraulic Fluids, AFWAL-TR-80-2111, Boeing Military Airplane Co., Seattle, WA, March 1982.

2. SAE AIR 1362, Aerospace Information Report, Physical Properties of Hydraulic Fluids, Society of Automotive Engineers, Inc., Warrendale, PA, May 1975.

3. Leo Parts, Assessment of the Flammability of Aircraft Hydraulic Fluids, AFAPL-TR-79-2055, Monsanto Research Corporation, Dayton, Ohio, July 1979.

4. T. L. Graham and W. E. Berner, Development of Seals for Nonflammable Hydraulic Fluids, AFML-TR-79-4143, Air Force Materials Laboratory, January 1980.

5. SAE ARP 24B, Aerospace Recommended Practice, Determination of Hydraulic Pressure Drop, Society of Automotive Engineers, Inc., Warrendale, PA, 1-31-68.

6. S. M. Warren and J. R. Kilner, Fireproof Brake Hydraulic System, AFWAL-TR-81-2080, Boeing Military Airplane Co., Seattle, WA, September 1981.

7. SAE AS 568A, Aeronautical Standard, Uniform Dash Number System for O-Rings, Society of Automotive Engineers, Inc., Warrendale, PA, July 1974.

8. Sperry Vickers Report, Qualification Test of Pump Model PV3-075-1 to MIL-P-19692B, Project No. 8-0115-204-340, February 29, 1968.

9. NAS 1638, National Aerospace Standard, Cleanliness Requirements of Parts Used in Hydraulic Systems, Aerospace Industries Association of America, Inc., Washington, D.C., January 1964.

APPENDIX A

NONFLAMMABLE AIRCRAFT HYDRAULIC FLUID SURVEY QUESTIONNAIRE AND ADDRESSEES

NON-FLAMMABLE AIRCRAFT HYDRAULIC FLUID SURVEY - QUESTIONNAIRE

NOTE: If you wish to propose more than one candidate fluid for consideration as a non-flammable hydraulic fluid for future Air Force aircraft, please reproduce the following pages of this questionnaire form and fill in a separate form for each fluid.

1. Does your company or agency have any materials having properties which meet the minimum requirements noted in Attachment A for candidate non-flammable hydraulic fluids which can be considered for possible use in future aircraft hydraulic systems?

   Yes______ No_____ Number of candidate fluids_________
   Fluids designations______________________________
   Fluid types or base______________________________

2. Do you know of any other companies or agencies which have or are developing such fluids? Yes_______ No________

   If yes, please note the company or agency name, address, and the name of the person to whom this survey should be directed.

If you are aware of any such activity in another company, division or subsidiary of your own corporation, please forward a copy of this questionnaire and cover letter directly to the recommended recipient(s). The questionnaire has already been sent to the following associated offices of your corporation/agency:

Please execute this questionnaire by August 1, 1976 and return to:

The Boeing Company
Attn: Mrs. Diane J. Fodor
BMAD Materiel Buyer, M. S. 41-13
P. O. Box 3999
Seattle, Washington 98124

Date________________ ____________________ ____________________
Reviewer's Name Title or Position

____
Company or Agency (complete title e.g. headquarters or specific division)

____
Complete Mailing Address

____
Phone Number (including Area Code)

NON-FLAMMABLE AIRCRAFT HYDRAULIC FLUID SURVEY - QUESTIONNAIRE

NOTE: When completing the following, feel free to add comments or modify questions, units, etc. for clarity. If data is not currently available but expected before January 1, 1977 write in the date. It is not necessary to have all values measured at this time in order to qualify as a candidate fluid.

* Please indicate test method
** Please indicate whether information is extrapolated or test data.

Your Fluid Designation ______________________________

Fluid Type or Base Stock ______________________________

FLAMMABILITY

Autogenous Ignition Temp. (AIT) ________ °F per * ______________

Flash Point ______________ °F per * ______________

Fire Point ______________ °F per * ______________

Hot Manifold Drip ______________ °F per * ______________

Hot Manifold Spray ______________ °F per * ______________

Atomized Fluid Flammability ________ °F per Attachment A

Heat of Combustion ______________ BTU/LBM per * ______________

PHYSICAL PROPERTIES

Density ____________ gm/cc at ________ °F**

____________ gm/cc at ________ °F**

____________ gm/cc at ________ °F**

____________ gm/cc at ________ °F**

Thermal Expansion Coefficient

____________ cc/cc/°F ________ °F**

____________ cc/cc/°F ________ °F**

Pour Point ____________ °F per * ______________________

Surface Tension ____________ dynes/cm@ __________ °F per* ______________

Fluid Designation

____

Viscosity, per* ____________________

Temperature	Viscosity Units
-65°F	__________ **
°F	__________ **
°F	__________ **
°F	__________ **
300°F	__________ **

Viscosity Index ____________________

Low Temperature Viscosity Stability per* ____________________

_____% change in viscosity after _____ hours at _______°F

Bulk Modulus per* ____________________

Bulk Modulus	Temperature	Pressure
________ psi at	________ °F	________ psi
________ psi at	________ °F	________ psi
________ psi at	________ °F	________ psi
________ psi at	________ °F	________ psi

Thermal Conductivity per* ____________________

________________BTU/HR/FT^2/FT at ________°F

________________BTU/HR/FT^2/FT at ________°F

Specific Heat per* ____________________

________________BTU/LB/°F at ________°F

________________BTU/LB/°F at ________°F

Vapor Pressure per* ____________________

________ mm Hg at ________°F

________ mm Hg at ________°F

________ mm Hg at ________°F

________ mm Hg at ________°F

Fluid Designation

____

Air Solubility per* ______________________________

______% volume at Atm. press. and ______ ^{o}F

______% volume at Atm. press. and ______ ^{o}F

Foaming Tendency per* ______________________________

Describe Results ______________________________

____

____

Electrical Conductivity

per* ______________________________

Lubricity

Shell 4 ball, __________ mat'l on __________ mat'l for __________ hr

__________ mm dia at ________ Kg at ______ ^{o}F at __________ rpm

__________ mm dia at ________ Kg at ______ ^{o}F at __________ rpm

__________ mm dia at ________ Kg at ______ ^{o}F at __________ rpm

__________ mm dia at ________ Kg at ______ ^{o}F at __________ rpm

Ryder Gear, ______________ mat'l for ________ hr.

________ ppi at __________ rpm at __________ ^{o}F at ______ ml/min flow

________ ppi at __________ rpm at __________ ^{o}F at ______ ml/min flow

CHEMICAL PROPERTIES

List Additives and Function (i.e. antifoaming agent, antioxident, etc.)

____

____

____

____

____

Neutralization Number per* ______________________________

________ mg KOH

Elastomer Compatibility

Material	Excellent	Fair	Poor	Incompatible
"L" Stock (Low Nitrile)	____	____	____	____
Other Buna N	____	____	____	____
Ethylene Propylene	____	____	____	____
Fluorocarbon	____	____	____	____
Other	____	____	____	____

Paint Compatibility

List compatible paint base types

____________________ to ________ °F
____________________ to ________
____________________ to ________
____________________ to ________

Metal/Element Compatibility

List Common Metal/Elements which are incompatible or should be avoided.

________ above ________ °F ________ above ________ °F
________ above ________ °F ________ above ________ °F
________ above ________ °F ________ above ________ °F

Electrical Insulation Compatibility

List compatible common electrical insulations

________ to ________ °F ________ to ________ °F
________ to ________ °F ________ to ________ °F

Toxicity Characteristics (Similar class materials information may be used)

Epidermal application ____________________
Single large dose ________ Repeated dosage ________
Oral ingestion ____________________
Eye irritation ____________________
Subcutaneous ingestion ____________________
Exposure to mist ____________________
Exposure to vapor ____________________
Pathological effects ____________________

Fluid Designation ______________________

ENDURANCE PROPERTIES

Thermal Stability (Nitrogen over fluid)

Test Method ______________________

Long Term Acceptability to ________ °F

Oxidative Stability (Air bubbled in, or over fluid)

Test Method ______________________

Long Term acceptability to ________ °F

Hydrolytic Stability (Nitrogen over fluid with 5% water)

Test Method ______________________

Long Term acceptability to ________ °F

Shear Stability per* ______________________

________ % reduction in viscosity at ______ °F

Valve Stiction per * ______________________

Long Term acceptability to ________ °F

DTA Incipient Oxidation Temperature per* ______________________

First exotherm at atm. press. ______ °F

AVAILABILITY (Checkfluids' production status)

________ Lab Produced ________ Pilot Plant ________ Full Scale Production

Delivery Time now	Delivery Time in 6 months	Quantity
________	________	3 gal
________	________	10 gal
________	________	50 gal

COST

Approximate $/Gal (1976 Dollars)

Now: $____________ for 3 gallons $________ for 10 gallons

$____________ for 50 gallons

1980-85 Production $ __________ at 10,000 gallons/year

1980-85 Production $ __________ at 100,000 gallons/year

1980-85 Production $ __________ at 1,000,000 gallons/year

ATTACHMENT A
TO SURVEY QUESTIONNAIRE

MINIMUM PROPERTY REQUIREMENTS FOR CANDIDATE NON-FLAMMABLE HYDRAULIC FLUIDS

In order to qualify for consideration as a non-flammable hydraulic fluid with capability for operation at system temperatures from -65 to 300F in future Air Force aircraft, candidate fluids must meet the following minimum requirements:

1. Chemical Structure:
   In order to meet the non-flammability requirements, there must be little or no hydrogen in the fluid composition. If the molecular structure of a basestock fluid is considered potentially satisfactory but appears incapable of being chemically tailored to provide the total requisite performance, suitable additives such as corrosion and/or oxidation inhibitors, metal deactivators, lubricity agents, viscosity index improvers, anti-foaming compounds, etc., which can be incorporated to stabilize or fortify the basic fluids should be identified.

2. Non-flammability per the following criteria:

Property	Test Method	Value
Autogenous Ignition Temperature	ASTM-D-2155 (Modified to include injection pressure to 1000 psig)	>1300F
Hot Manifold Ignition Temperature	Modified Federal Test Standard 791B - Method 6053	>1700F
Heat of Combustion	ASTM D-240 (Bomb Method)	<5000 BTU/LbM
Atomized Fluid Flammability Test	AFAPL Procedure as follows:	self extinguishing flame

The fluid shall be sprayed through a 70° spray cone, 2.25 gallons per hour oil burner nozzle (used in home oil burners) that has been drilled out to .016" diameter. The nozzle pressure and temperature shall be 300 psig and 65± 10F, respectively. The system shall be capable of delivering fluid at these conditions for at least three minutes after the ignition has been applied. The ignition sources shall be (1) 6-joule 20K volt spark, (2) 6-inch high pre-mixed, stoichiometric propane-air flame emanating from 3/4" I.D. burner, and (3) incendiary

gunfire simulator (AFAPL-TR-73-50, Incendiary Gunfire Simulation Techniques for Fuel Tank Explosion Protection Testing). These ignition sources shall be two feet from the nozzle and intersecting its spray centerline. The success criteria for the spark ignition source and the simulated incendiary gunfire shall be that the fluid may flash, however, the flame must be self-extinguishing. The success criteria for the propane-air flame ignition source shall be that the fluids flame front does not propagate back to the nozzle and that the flame must be self-extinguishing when the ignition source is removed.

Available test data will be accepted for initial evaluation provided that the test methods are identical. Fluid samples will be submitted to the AFAPL for final flammability testing.

3. Pour Point

To be usable at the specified minimum system operating temperature of -65F, the fluid pour point should be below -75F.

4. Viscosity

To be usable throughout the specified system operating temperature range of -65F to 300F, a fluid viscosity no greater than 2500 centistokes at -65F and no less than 2 centistokes at 300F is desired. Exceptions will be considered provided the fluid is considered usable throughout the specified temperature range without entailing undue penalties in system design and/or operational performance.

5. Thermal Stability

Stability at 300F in the presence of typical system metals is required. Data indicating high temperature instability, such as evidence of undue change in a fluid's viscosity or increase in acid number, or in weight loss or other evidence of corrosive attack on typical system materials could be cause for rejection. It is desired that each viable candidate be stable in the presence of all the following materials, but some exceptions may be allowed if material substitutions can be made without undue penalty:

bare carbon steels, stainless steels, bearing steels, aluminum, beryllium copper, bronze, and titanium alloys, chrome, cadmium, electro-less nickel, and silver platings.

6. Toxicity

Inasmuch as aircraft hydraulic fluids must be safely handled by main-

tenance personnel without protective equipment, it is required that no health hazard or cumulative toxic effect result from skin contact with any fluid proposed or from the breathing of its vapors under normal handling conditions. Available test data and values for similar chemical types will be accepted for initial evaluation.

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APPENDIX B

THE REDUCTION IN HYDRAULIC TUBE SIZES AND WEIGHT OBTAINABLE WITH A LOWER VISCOSITY CTFE FLUID

As noted in Subsection 2.3.6, one method for reducing the weight penalty which would be incurred by designing a hydraulic system for use with the AO-8 CTFE fluid is to use a lower viscosity blend of the fluid in order to reduce tubing sizes and fluid volume. As shown in Figure 2, the AO-8 fluid has a viscosity in the normal operating temperature range similar to that of MIL-H-5606 fluid. This was obtained by adding a viscosity-index improver additive to the basestock fluid: the Halocarbon Products' 1.8/100 fluid which has a viscosity approximately one-sixth that of the AO-8 fluid at design temperatures around 50F. (See Figure 5 in the Design Guide, Reference 1.)

Since the thickness of hydrodynamic lubrication films and their load carrying ability are a direct function of the lubricating fluid's viscosity, units such as hydraulic pumps and motors which are lubricated by the hydraulic fluid must be designed with bearing areas large enough to sustain all operating loads with the fluid viscosity existent over the full range of operating temperatures. If hydraulic pumps and motors can be designed to operate satisfactorily with the 1.8/100 fluid, it could be considered a viable choice; and, system weight could be reduced since its lower viscosity would allow smaller size tubing than that required for CTFE fluids, such as AO-8, with viscosities comparable to hydraulic fluids currently in use.

Tube Sizing Equations

The following equations are taken from Eq. (29) and Eq. (31) shown in Subsection 2.3.4.1.1 for flow losses for laminar flow and turbulent flow respectively.

For laminar flow: $D = \left(\dfrac{s\nu QL}{3663\Delta P}\right)^{1/4}$ . . . . (B1)

For turbulent flow: $$D = \left(\frac{s\nu^{0.25}Q^{1.75}L}{1756\Delta P}\right)^{1/4.75} \qquad \text{(B2)}$$

where:

- D = Tube inside diameter (in)
- s = Fluid specific gravity, dimensionless, or Fluid density, (g/cm^3)
- ν = Fluid kinematic viscosity (cs)
- Q = Fluid flow rate (gpm)
- L = Length of tube (ft)
- ΔP = Pressure loss (psi)

To determine which equation to use, it is necessary to first calculate the Reynolds number. Any one of the four equations noted in 2.3.4.1.1, Eq. (20), Eq. (21), Eq. (22), or Eq. (23), may be used depending upon the fluid flow parameters which are known, or can be estimated, at the time.

When attempting to study the weight savings due to the utilization of a lower viscosity fluid, one must consider that, with increasing flow, each fluid will pass from laminar to turbulent flow at a different flow rate. This can be seen by noting that the formulae for Reynolds number include both fluid density and viscosity.

Considering the case where the flow rate of two fluids are increased equally from zero such that both fluid flows start out as laminar, eventually one fluid will break into turbulent and then the other fluid will go turbulent. Since hydraulic lines are of specific sizes (i.e. 3/8, 1/2, 5/8, etc), a system will have several design flow rates for each diameter. For this reason, the study must consider a variable flow rate for the size ratioing.

The plumbing diameter equations for a reference fluid (subscript 1) and a study fluid (subscript 2) for all possible flow conditions are as follows:

Flow Conditions
(reference fluid
to study fluid)

laminar to laminar
$$D_2 = \left(\frac{s_2}{s_1}\frac{\nu_2}{\nu_1}\right)^{1/4} D_1 \quad \text{(B3)}$$

laminar to turbulent
$$D_2 = \left[\left(\frac{\nu_2}{\nu_1{}^4}\right) Q^3 \left(\frac{3663}{1756}\right)^4 D_1^{16} \left(\frac{s_2}{s_1}\right)^4\right]^{1/19} \quad \text{(B4)}$$

turbulent to laminar
$$D_2 = \left[\left(\frac{1756}{3663}\right)^4 \left(\frac{D_1}{Q^3}\right)^{19} \left(\frac{s_2}{s_1}\right)^4 \frac{\nu_2{}^4}{\nu_1}\right]^{1/16} \quad \text{(B5)}$$

turbulent to turbulent
$$D_2 = D_1 \left[\left(\frac{s_2}{s_1}\right)^4 \frac{\nu_2}{\nu_1}\right]^{1/19} \quad \text{(B6)}$$

When either the fluid viscosity and/or density is reduced significantly, the calculated equivalent tube diameter to attain the same pressure loss per unit of line length is reduced. With the same flow rate in the smaller diameter tube, the fluid velocity will be greater. However, care must be taken that velocities do not exceed values which will cause the pressure rise resulting from abrupt valve closure to exceed the 35% limit specified in MIL-H-5440.

In order to determine the magnitude of weight reduction which could be realized with a lower viscosity CTFE fluid, a study of the Air Force/ Boeing YC-14 Advanced Medium STOL Transport (AMST) aircraft was made. Both the increase in line weight, which would result from the increased tube diameters (and higher fluid density) required to convert the existing hydraulic system from MIL-H-5606 fluid to AO-8 CTFE fluid, and the weight reduction obtainable through use of the 1.8/100 CTFE fluid in lieu of the AO-8 fluid, were estimated.

For that study, the following parameters were used to calculate the

fluid velocities which would cause a 35% pressure rise (1,050 psi in a 3,000-psi system) due to abrupt valve closure. Note that the assumed fluid/system compliance values are lower than the measured bulk modulus values in order to account for the typical fluid condition with entrained air.

Parameter	MIL-H-5606 Fluid	AO-8 and 1.8/100 CTFE Fluids
Density (at Room Temperature)	0.85 g/cm^3	1.836 g/cm^3
Adiabatic-Tangent Bulk Modulus (R.T. @ 3,000 psi)	286,000 psi	245,000 psi
Assumed Fluid/System Compliance	150,000 psi	125,000 psi
Fluid Velocity for 1,050-psi Pressure Rise due to sudden Valve Closure	25 fps	18.8 fps

In order to minimize computation time, an interactive computer program was written to accomplish the necessary calculations. As written for the YC-14 system study, the program incorporates the weight calculations for tube wall thickness for 21Cr-6Ni-9Mn alloy stainless steel pressure lines for a 3,000-psi system (12,000-psi minimum pressure burst requirement), and 6061-T6 aluminum return lines for design pressures from 600 psi to 1,500 psi (1,800-psi and 4,500-psi minimum burst pressure requirement) depending upon tube diameter. The weight figures are for tubing full of fluid (wet weight) and include a 20% allowance (of tubing dry weight) for end fittings and tube nuts and a 10% allowance (of the tubing wet weight) for tube support clamps.

For that airplane, the minimum hydraulic fluid full-flow design temperature is +50F which is representative of military transports and commercial airliners which are allowed warmup periods whenever they are cold soaked t lower temperatures. Systems for other aircraft, such as all-weather fighters or strategic bombers on ready alert status, must be designed to deliver high flows at considerably lower (subzero) temperatures. For those aircraft the relative system weight penalties to accommodate CTFE fluid will be higher.

Results

The results of the YC-14 study are shown in the following Table B1. The existing pressure and return line diameters and wall gages are tabulated in the first column. The total lengths of each tube size are tabulated in the second column. The installed wet weights of tubing, fittings, and clamps for each tube size in the existing MIL-H-5606 fluid system are tabulated in the third column. In the fourth column, both the weight ratios for the AO-8 CTFE fluid tubing to MIL-H-5606 fluid tubing, and the installeld weight of tubing, fittings, and clamps for each tube size required for an AO-8 CTFE fluid system are tabulated. In the fifth column, both the weight ratios for the 1.8/100 fluid tubing to MIL-H-5606 fluid tubing, and the installed weight of tubing, fittings, and clamps for each tube size required for a 1.8/100 CTFE fluid system are tabulated.

As seen at the bottom of the tabulations in the third column, the total weight of the MIL-H-5606 fluid plumbing system is 1,538.4 lb. As seen at the bottom of the fourth column, the total weight of an AO-8 CTFE fluid plumbing system would be 3,087.6 lb which represents an increase of 1,549.2 lb (100.7%) over the existing system. As seen at the bottom of the fifth column, the total weight of a 1.8/100 CTFE fluid plumbing system would be 2,207.3 lb which represents an increase of 668.9 lb (43.5%) over the existing (MIL-H-5606 fluid) plumbing system but a decrease of 880.3 lb (28.5%) from an AO-8 CTFE fluid plumbing system.

TABLE B.1 YC-14 AMST HYDRAULIC TUBING SYSTEM WEIGHTS FOR MIL-H-5606 FLUID, AO-8 CTFE FLUID, AND 1.8/100 CTFE FLUID

Tube Diameter and Wall Thickness	Tube Length (feet)	MIL-H-5606 Installed Wet Weight (pounds)	$\frac{W_{AO-8}}{W_{5606}}$	AO-8 CTFE Installed Wet Weight (pounds)	$\frac{W_{1.8/100}}{W_{5606}}$	1.8/100 Installed Wet Weight (pounds)
PRESSURE LINES						
3/8 x .020	1016	134.5	1.63	219.2	1.25	168.1
1/2 x .026	494	116.9	1.72	201.1	1.42	166.0
5/8 x .033	302	111.3	1.72	191.4	1.42	158.0
3/4 x .039	342	182.2	1.72	313.4	1.42	258.7
1 x .052	409	387.3	1.72	666.2	1.42	550.0
		932.2 lb.		1591.3 lb.		1300.8 lb.
RETURN LINES						
3/8 x .035	395	34.3	1.70	58.3	1.49	51.1
1/2 x .035	426	58.4	1.85	108.0	1.60	93.4
5.8 x .035	395	78.0	1.97	153.7	1.70	132.6
3/4 x .035	187	50.1	2.06	103.2	1.77	88.7
1 x .035	287	125.8	2.19	275.5	1.87	235.2
1-1/4 x .035	112	72.6	2.28	165.5	1.95	141.6
1-1/2 x .035	208	187.0	3.38	632.1	0.88	163.9
		606.2 lb.		1496.3 lb.		906.5 lb.
TOTAL WEIGHT		1538.4 lb.		3087.6 lb.		2207.3 lb.
Weight Change Relative to 5606				+1549.2 lb.		+668.9 lb.
		Weight Change Relative to AO-8				-880.3 lb.

APPENDIX C
FIFTY-HOUR PUMP TEST PLAN

The Fifty-Hour Pump Test outlined herein will meet the requirements specified in Section F, paragraph 4.3.3 of the subject contract. The selected pump is a Vickers PV3-075-15 with a rated speed of 7000 rpm. The system fluid will be Halocarbon AO-8. The pump operating characteristics and expected life will be determined with the AO-8 fluid. Pump teardown inspection will be accomplished to determine required design mmodificaitons. Testing will be halted for any sudden performance degradation and a teardown inspection performed.

All testing will be performed in a test setup as shown in Figure B1. The reservoir pressure unless otherwise specified will be a value to produce 100 psia at the pump inlet at rated speed and rated flow. The compensator will be adjusted to 3025 $\pm$25 psi at 120F and zero flow, and any change in compensator setting throughout the tests will be noted. The shaft seal leakage will be monitored and recorded throughout the testing.

Inspection

A teardown inspection will be performed before testing, between endurance phases and at the completion of testing. Piston shoe clearance and wear patterns will be recorded and compared. The test system filters will be inspected at the end of testing for pump wear particles and fluid breakdown products.

Performance Tests

The pump will be performance tested at three speeds and three inlet temperatures. The speeds will be 3500, 5250, and 7000 rpm. The inlet temperatures will be 120F, 180F, and 240F. Each of the tests will be performed as follows:

a. Set varidrive speed
b. Stabilize inlet temperature
c. Adjust discharge flow to 0 gpm
d. Wait two minutes and record the following data:
    1. Pump input speed

2. Pump input torque
3. Discharge flow
4. Case drain flow
5. Suction pressure
6. Discharge pressure
7. Case drain pressure
8. Suction temperature
9. Discharge temperature
10. Case drain temperature

e. Increase discharge flow by 2 gpm and repeat d & e until reaching full rated flow
f. Repeat d at 2500, 1500, 1000, and 500 psi discharge pressures.

The data obtained from the nine performance tests will be analyzed for efficiency, heat rejection and delivery characteristics. The final data will be in the form of graphs.

Critical Inlet Tests

Critical inlet tests will be performed at three speeds and three inlet temperatures. The speeds and inlet temperatures will be the same as in the performance test. Each of the tests will be performed as follows:

a. Set varidrive speed
b. Stabilize inlet temperature
c. Adjust discharge pressure to maximum full-flow pressure and maintain this setting throughout test
d. Decrease pump inlet pressure in 2-psi increments starting at 60 psia and record the following data. Reduce the inlet pressure until the discharge flow has decreased to 50 percent of the flow at 60 psia.
   1. Discharge flow
   2. Suction pressure
   3. Discharge pressure
   4. Pump speed
   5. Inlet temperature

The data obtained from the nine critical inlet tests will be plotted to determine the critical inlet characteristics of the pump.

Maximum Pressure, Response Time, and Pressure Pulsation Tests

These tests will be run in accordance with paragraph 4.3.2.1.5 through 4.3.2.1.5.4 of MIL-P-19692C with the following additional requirements:

1. The tests will be run with two different fluid compliance volumes, one with a pressure volume of approximately 50 cu.in. and the other with a pressure volume of approximately 250 cu.in.
2. The tests will also be run with two different fluid inlet temperatures: 120F and 240F.

Endurance Time

The remaining time of the fifty-hour pump test will be an endurance test, half consisting of a normal-rated test phase and half an overload test phase. The normal-rated test phase will consist of half time each at conditions of Phases 6 and 7, Table IV of MIL-P-19692C. Following this test phase, the pump will undergo a teardown inspection for wear determination. The pump will then be reassembled, the disturbed seals replaced, and a break-in run conducted per Component Maintenance Manual 29-10-09, pages 701-708 (Vickers Manual Number 910269 dated 15 July 1972). The overload test phase will then be run per the conditions of Phase 7, Table V of MIL-P-19692C for the remaining time.

Test Condition Tolerances

Test conditions will be held as close to the stated values as is practical with the existing test equipment. The following tolerances are typical.

Regulated Temperatures:	±5F
High Pressures:	±50 psi
Low Pressures:	±5 psi

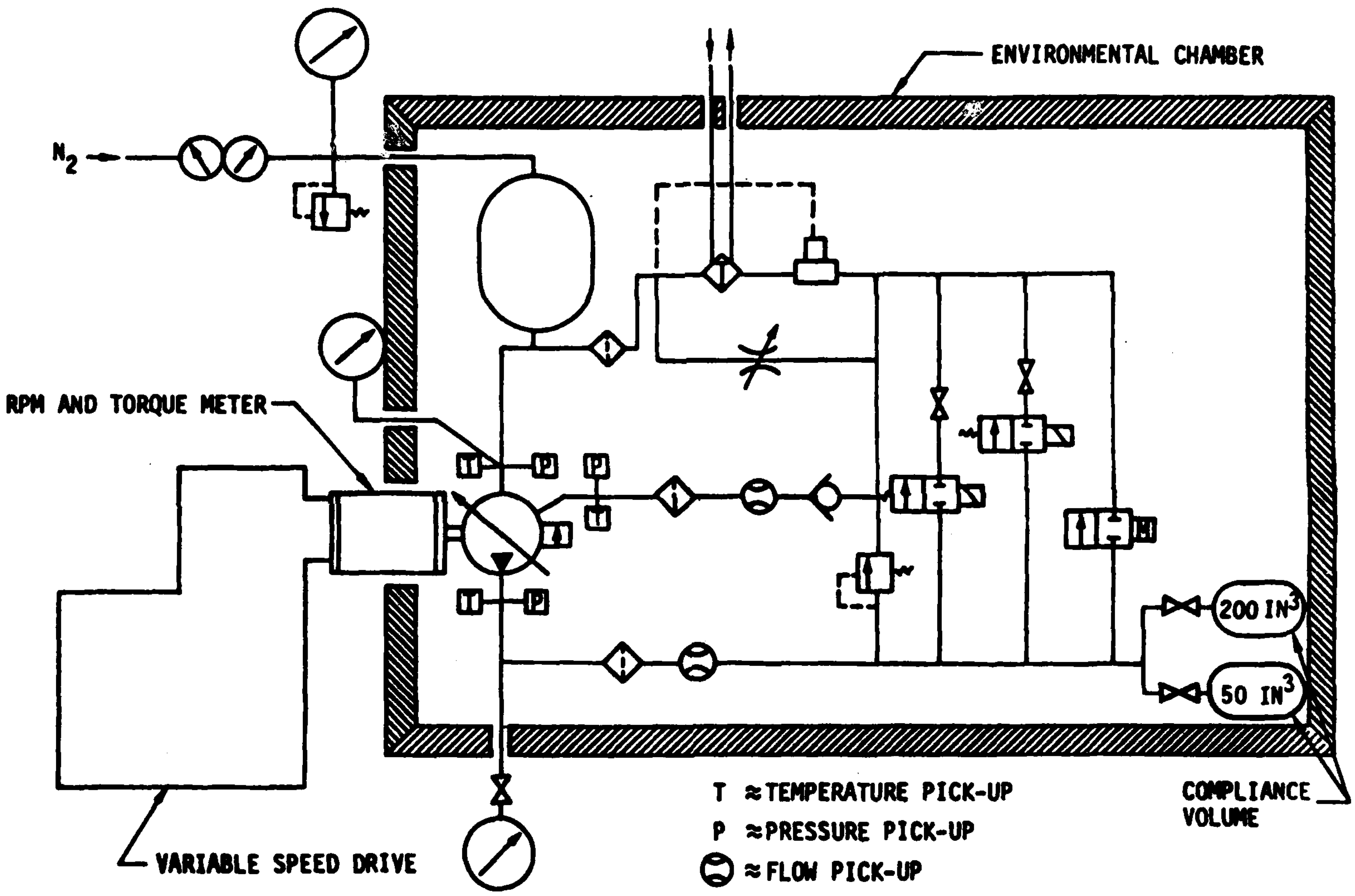
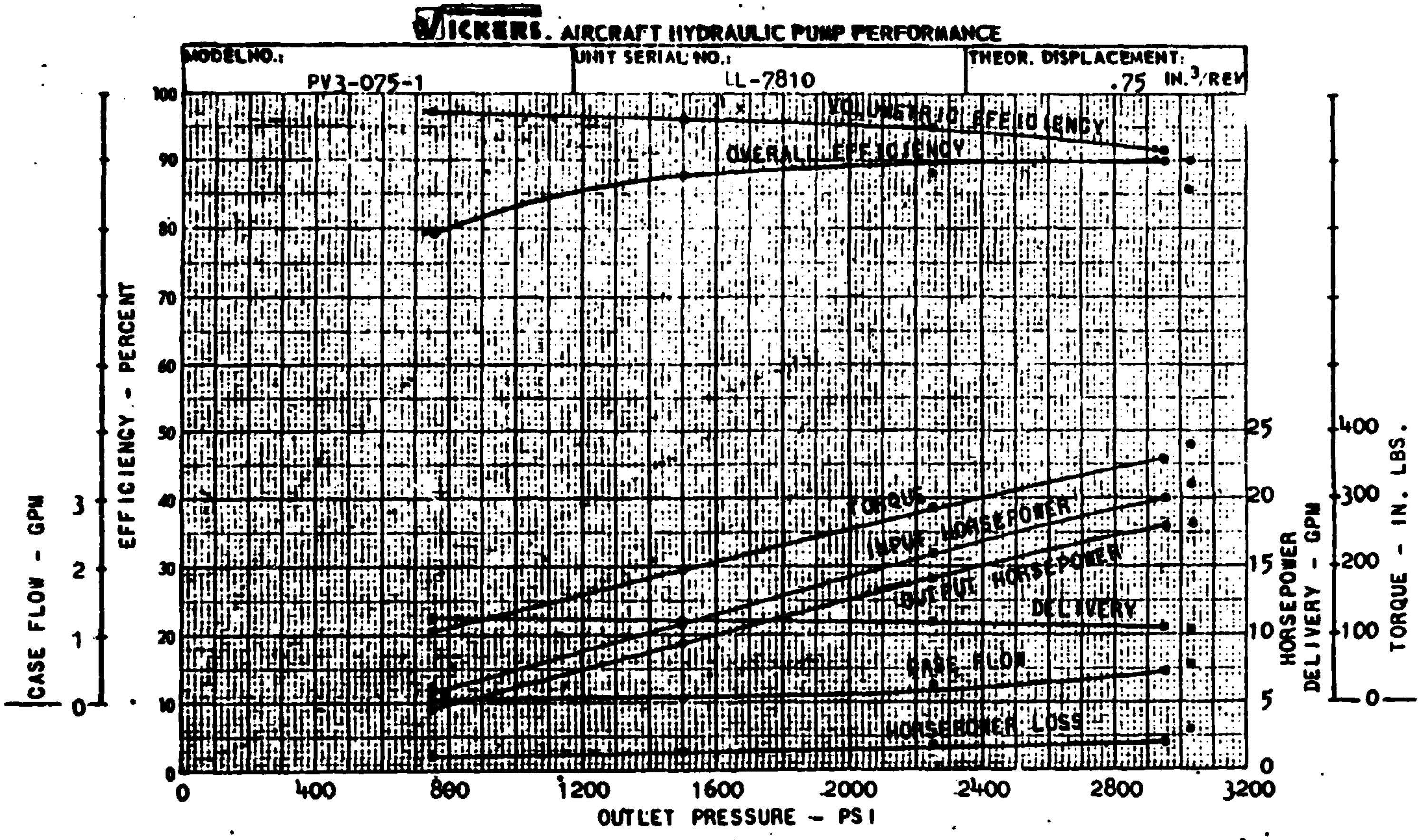


Figure C1. Fifty-hour pump test setup

APPENDIX D

EXCERPT PAGES FROM SPERRY-VICKERS REPORT
QUALIFICATION TEST OF PUMP MODEL PV3-075-1 TO MIL-P-19692B,
PROJECT NO. 8-0115-204-340 DATED 2-29-68



SPEED: 3500 RPM
TEMPERATURE: 240 ± 10°F
FLUID: MIL-H-5606B
INLET PRESSURE: 40 PSIG
OUTLET PRESSURE: AS NOTED PSIG
CASE PRESSURE: 55 PSIG
BAROMETER: 29.40 IN. HG.
CALIBRATED DISPL.: .7611 IN.³/REV

REMARKS:
DELIVERY IS BASED UPON EXPANDED VOLUME (ZERO PRESSURE)
INITIAL CALIBRATION PER MIL-P-19692B PAR 4.3.2.1.4

DATA SHEETS:
8-0115-204-340-245-2

NOTE: These data reflect performance of one pump, and must not be considered representative of average or guaranteed values for production pumps.

CURVE NO. 8-0115-204-108C
TEST DATE 4-22-67
APPROVAL
1.
2. 670512
3.

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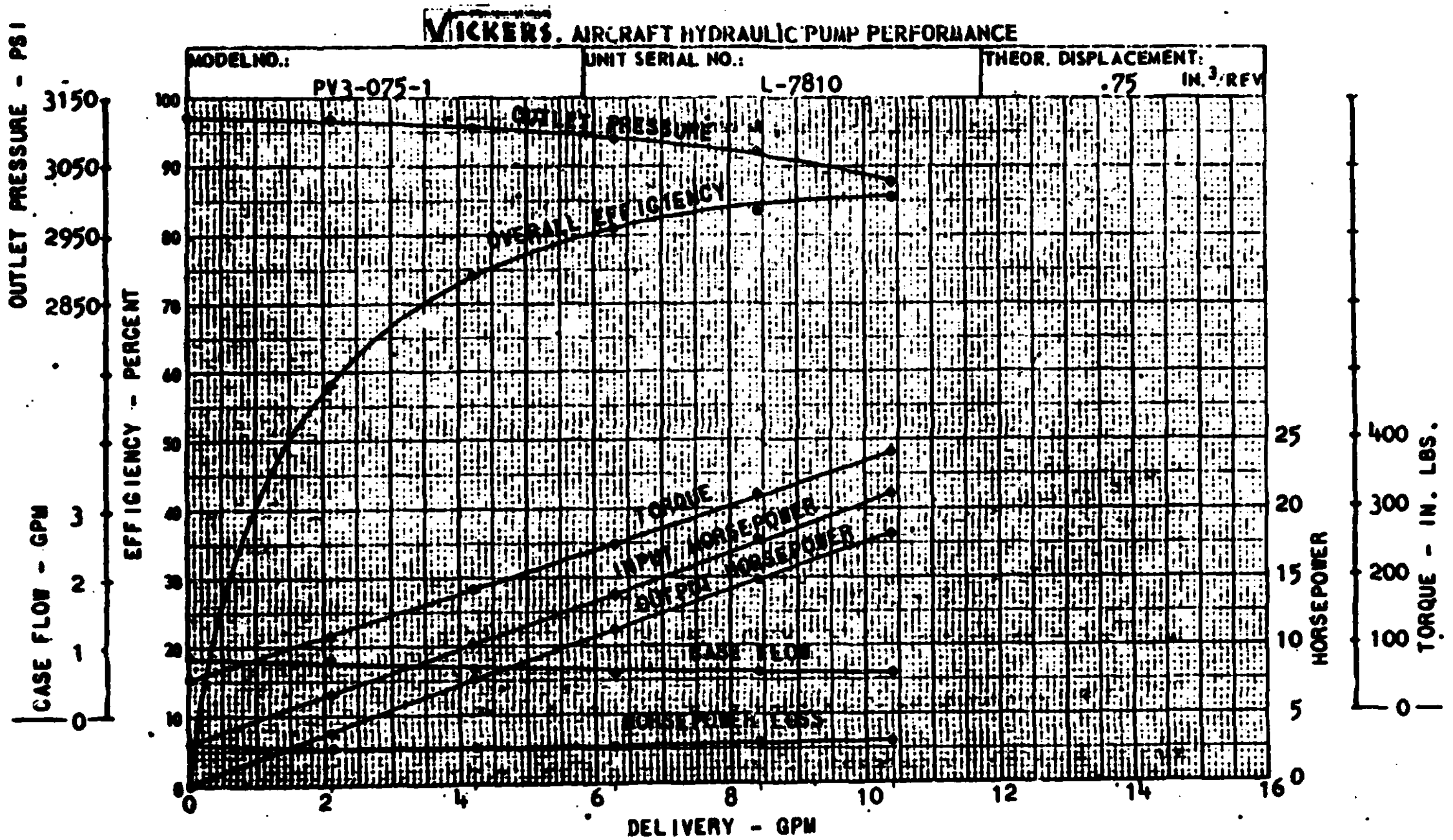
Figure D-1 Qualification pump full-flow performance at 3500 rpm



SPEED: 3500 RPM TEMPERATURE: 240 ± 10°F FLUID: MIL-H-5606B INLET PRESSURE: 40 PSIG OUTLET PRESSURE: AS NOTED PSIG CASE PRESSURE: 55 PSIG BAROMETER: 29.40 IN. HG. CALIBRATED DISPL.: .7611 IN.3/REV	REMARKS DELIVERY IS BASED UPON EXPANDED VOLUME (ZERO PRESSURE) INITIAL CALIBRATION PER MIL-P-19692B PAR 4.3.2.1.4 DATA SHEETS: 8-0115-204-340-245-2 NOTE: These data reflect performance of one pump, and must not be considered representative of average or guaranteed values for production pumps.	CURVE NO. 8-0115-204-108D TEST DATE 4-22-67 APPROVAL 1. 2. HH 670512 3.

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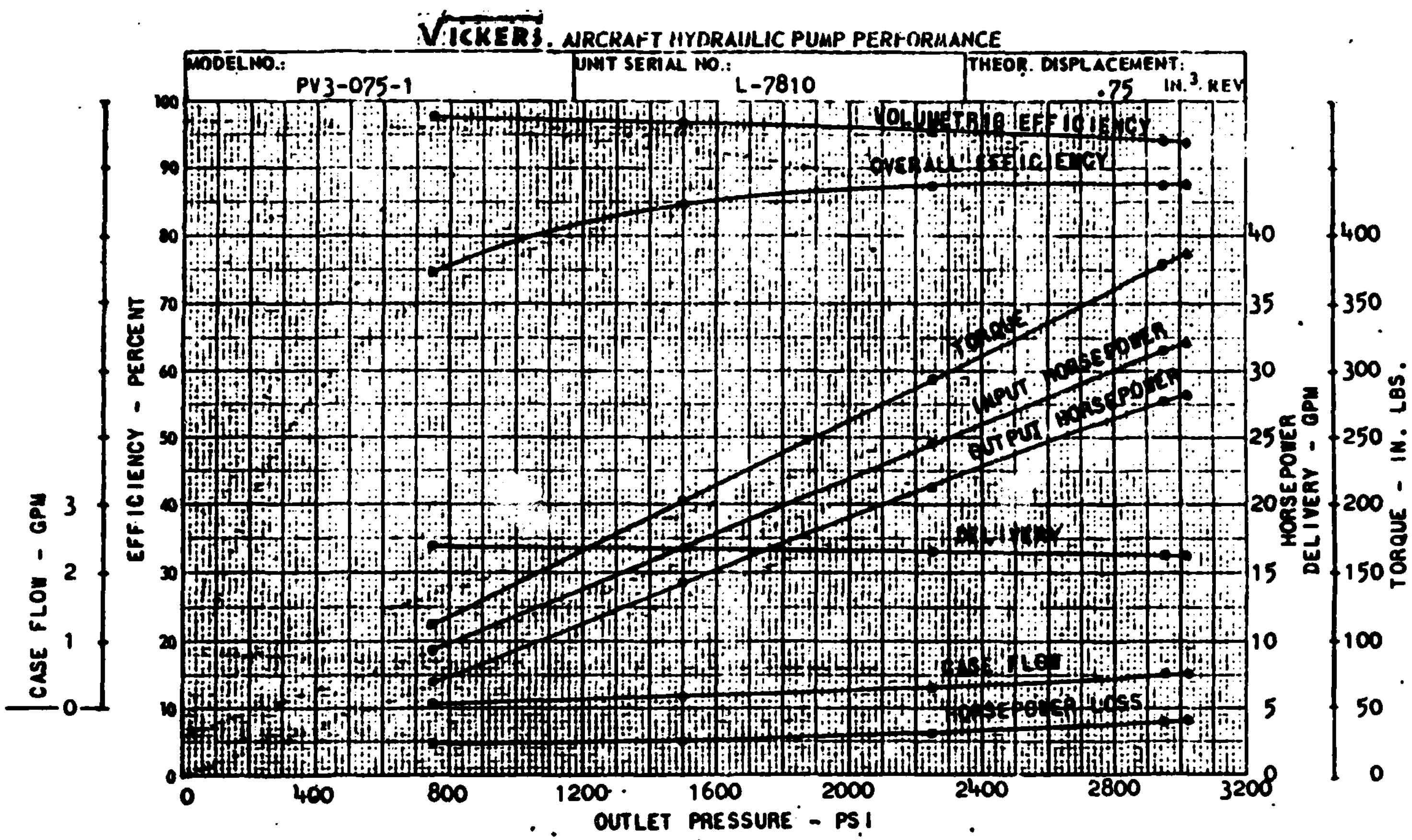
Figure D-2 Qualification pump partial-flow performance at 3500 rpm



SPEED: 5250 RPM TEMPERATURE: 240 ± 10°F FLUID: MIL-H-56068 INLET PRESSURE: 40 PSIG OUTLET PRESSURE: AS NOTED PSIG CASE PRESSURE: 55 PSIG BAROMETER: 29.40 IN. HG. CALIBRATED DISPL.: .7611 IN.3/REV	REMARKS- DELIVERY IS BASED UPON EXPANDED VOLUME (ZERO PRESSURE) INITIAL CALIBRATION PER MIL-P-19692B PAR 4.3.2.1.4 DATA SHEETS: 8-0115-204-340-245-3 NOTE: These data reflect performance of one pump, and must not be considered representative of average or guaranteed values for production pumps.	CURVE NO. 8-0115-204-108E TEST DATE 4-22-67 APPROVAL 1. 2. *HK 670512* 3.

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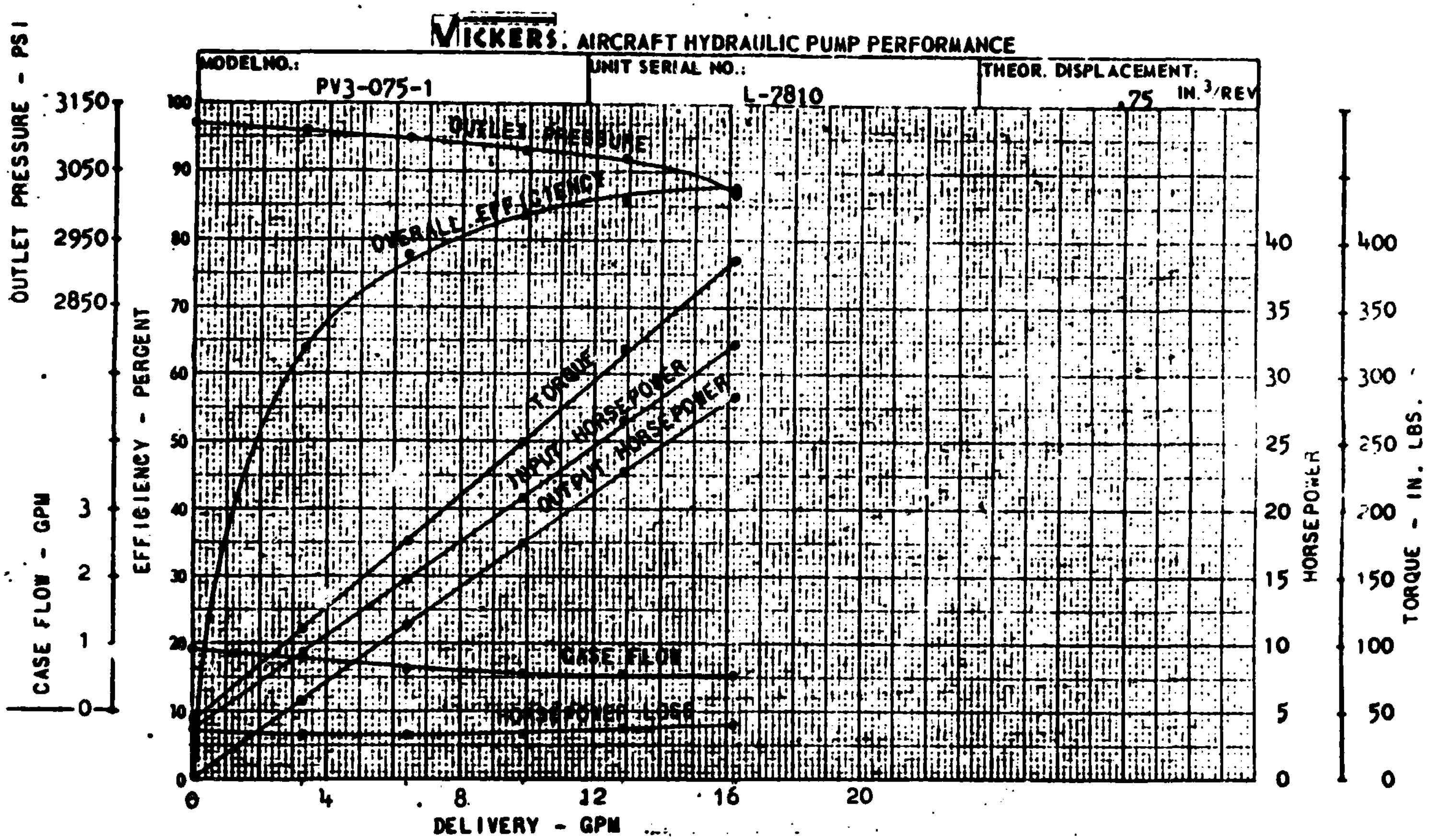
Figure D-3 Qualification pump full-flow performance at 5250 rpm



SPEED: 5250 RPM TEMPERATURE: 240 ± 10°F FLUID: MIL-H-5606B INLET PRESSURE: 40 PSIG OUTLET PRESSURE: AS NOTED PSIG CASE PRESSURE: 55 PSIG BAROMETER: 29.40 IN. HG. CALIBRATED DISPL.: .7611 IN.³/REV	REMARKS DELIVERY IS BASED UPON EXPANDED VOLUME (ZERO PRESSURE) INITIAL CALIBRATION PER MIL-P-19692B PAR 4.3.2.1.4		CURVE NO. 8-0115-204-108F TEST DATE 4-22-67 APPROVAL 1. 2. WH 670512 3.
	DATA SHEETS: 8-0115-204-340-245-3	NOTE: These data reflect performance of one pump, and must not be considered representative of average or guaranteed values for production pumps.	

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Figure D-4 Qualification pump partial-flow performance at 5250 rpm

AD-A118 169 BOEING MILITARY AIRPLANE CO SEATTLE WA F/G 11/8

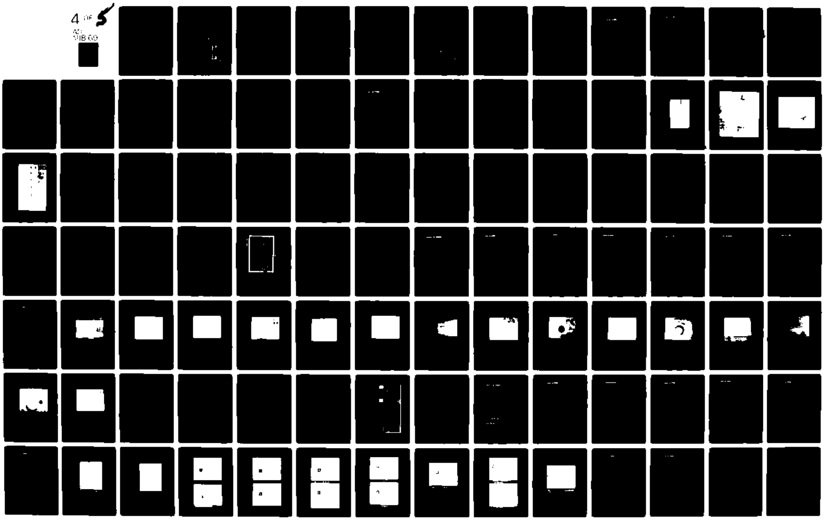
FIRE RESISTANT AIRCRAFT HYDRAULIC SYSTEM.(U)

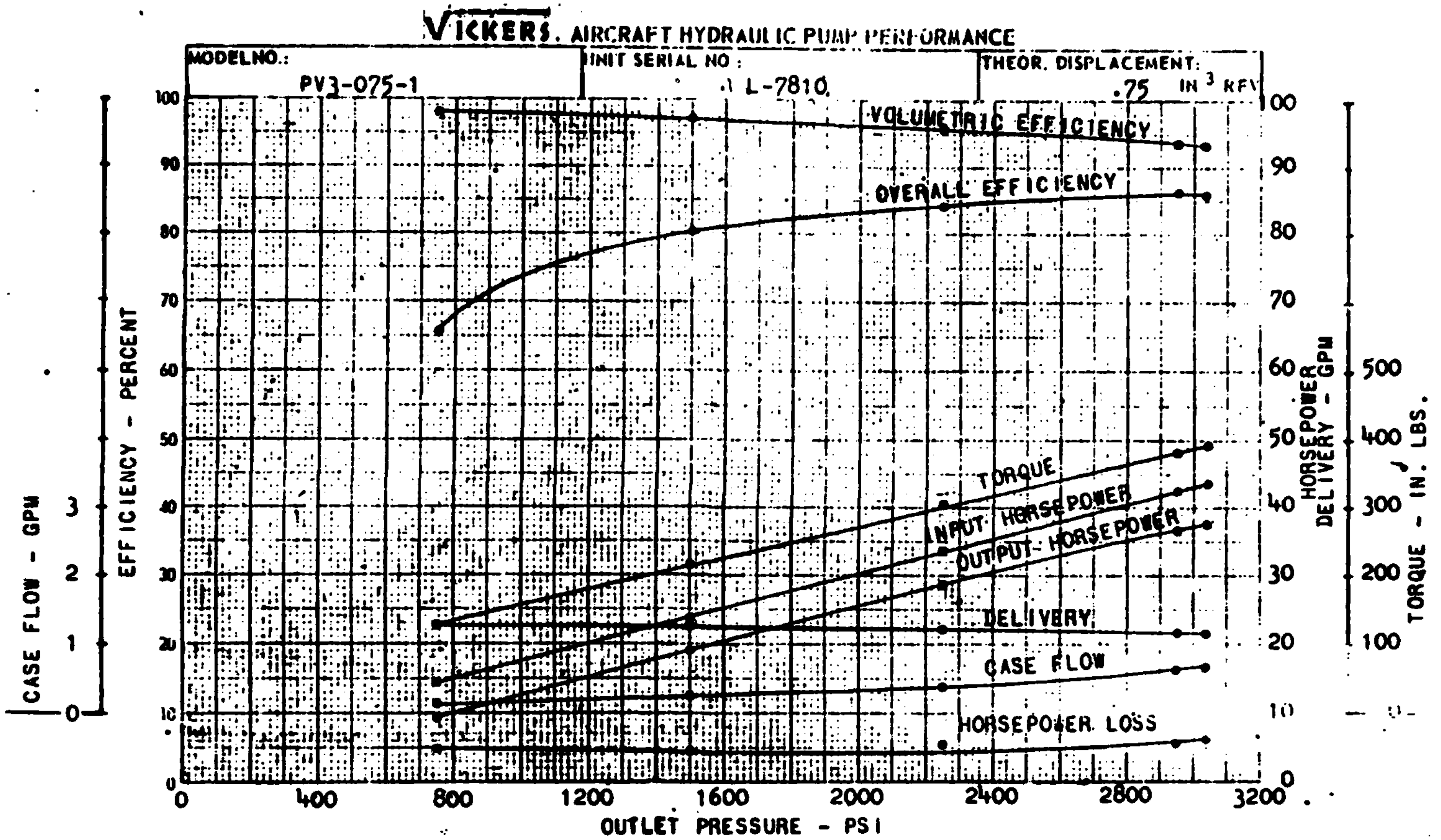
JUL 82 E T RAYMOND, D W HULING, R L SHICK F33615-76-C-2064

UNCLASSIFIED AFWAL-TR-80-2112 NL

4 OF 5

AD A118169





SPEED: 7000 RPM TEMPERATURE: 240 ± 10°F FLUID: MIL-H-5606B INLET PRESSURE: 45 PSIG OUTLET PRESSURE: AS NOTED PSIG CASE PRESSURE: 60 PSIG BAROMETER: 29.40 IN. HG. CALIBRATED DISPL.: .7611 IN.³/REV	REMARKS: DELIVERY IS BASED UPON EXPANDED VOLUME (ZERO PRESSURE) INITIAL CALIBRATION PER MIL-P-19692B, PAR 4.3.2.1.4 DATA SHEETS: 8-0115-204-340-245-4 NOTE: These data reflect performance of one pump, and must not be considered representative of average or guaranteed values for production pumps.	CURVE NO. 8-0115-204-108G TEST DATE. 4-22-67 APPROVAL 1. 2. 670572 3.

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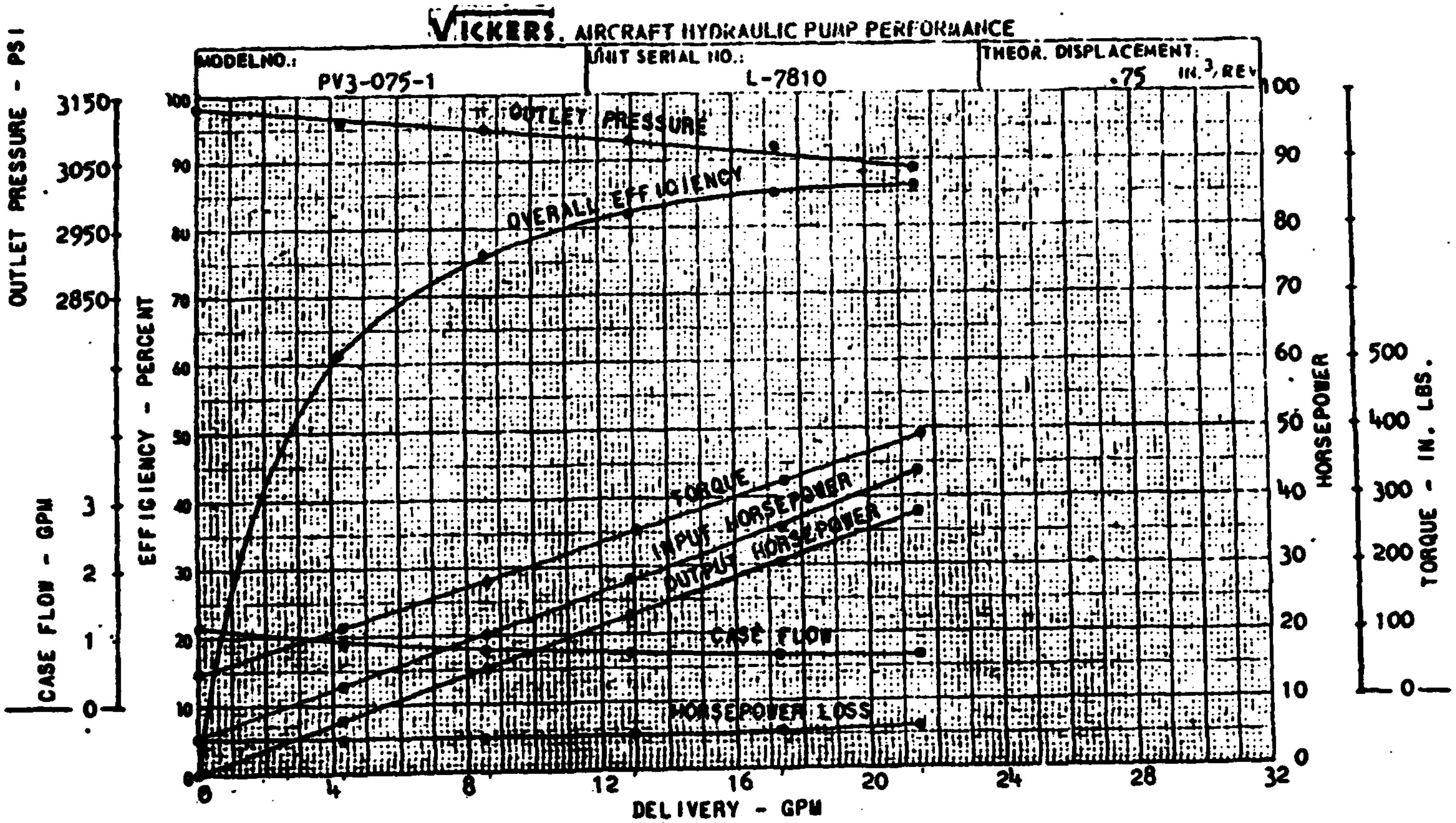
Figure D-5 Qualification pump full-flow performance at 7000 rpm



SPEED: 7000 RPM
TEMPERATURE: 240 ± 10°F
FLUID: MIL-H-56068
INLET PRESSURE: 45 PSIG
OUTLET PRESSURE: AS NOTED PSIG
CASE PRESSURE: 60 PSIG
BAROMETER: 29.40 IN. HG.
CALIBRATED DISPL.: .7611 IN.^{3}/REV

REMARKS:
DELIVERY IS BASED UPON EXPANDED VOLUME.
(ZERO PRESSURE)

INITIAL CALIBRATION PER MIL-P-19692B PAR 4.3.2.1.4

DATA SHEETS: 8-0115-204-340-245-4

NOTE: These data reflect performance of one pump, and must not be considered representative of average or guaranteed values for production pumps.

CURVE NO. 8-0115-204-108H
TEST DATE
APPROVAL
1.
2. 670512
3.

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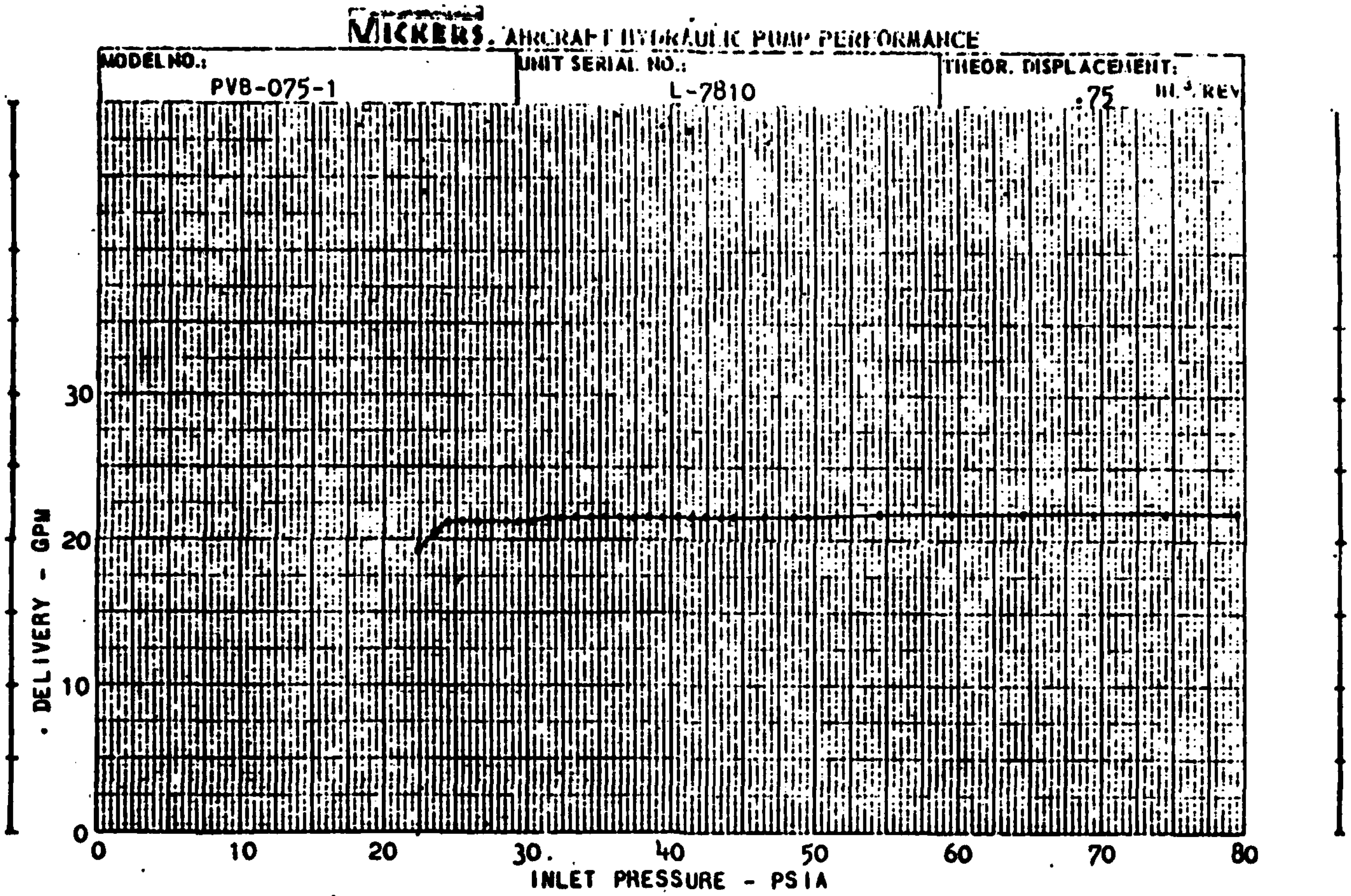
Figure D-6 Qualification pump partial-flow performance at 7000 rpm



SPEED: 7000 RPM
TEMPERATURE: 240 ± 10°F
FLUID: MIL-H-5606B
INLET PRESSURE: AS NOTED
OUTLET PRESSURE: 2950 PSIG
CASE PRESSURE: 62 PSIG
BAROMETER: 29.22 IN. HG.
CALIBRATED DISPL.: .7611 IN.³/REV

REMARKS: DELIVERY IS BASED UPON EXPANDED VOLUME (ZERO PRESSURE)

Cavitation Test

DATA SHEETS: 8-0115-204-340-273

NOTE: These data reflect performance of one pump, and must not be considered representative of average or guaranteed values for production pumps.

CURVE NO. 8-0115-204-113
TEST DATE 8-3-67
APPROVAL
1. R. J. D. 10-26-67
2. HH 671026
3.

Figure D-7 Qualification pump critical inlet pressure at 7000 rpm

PUMP SPEED RPM	BRMC IN HG.	PRESSURE PSIG PI	PO	PC	FLOW GPM	% OF FLOW	TEMP °F. TI	TO	TC	Response Time Seconds 90-5%	5-90%	TAPE #
3500	29.3	40	2950	55	10.53	100	243	252	262			
3492		40	3060	57	9.51	90	235	243	263	.0156	.046	1
3504		50	3125	67	.6	5	217	230	252			
5258		40	2950	55	16.77	100	235	243	270			
5253		40	2950	74	14.69	90	235	243	270	.0136	.035	2
5265		60	3130	56	.87	5	215	235	260			
7000		45	2950	60	21.6	100	245	255	288			
6999		45	3040	60	19.47	90	242	252	283	.0[illegible]0	.0[illegible]0	3
7010		70	3145	75	1.22	5	230	243	268			

Figure D-8 Qualification pump response times per MIL-P-19692B

PHOTO NO.	PUMP SPEED RPM	OUTLET PRESSURE PSIG		DELIVERY		TEMP. °F. INLET	PULSATION PEAKS STEADY STATE
		#1	#2	GPM	%		PEAK TO PEAK
1A	3500	2950	1475	10.45	100	240	225 PSI
2A	3500	3080	1540	7.80	75	230	180
3A	3496	3090	1545	5.2	50	245	275
4A	3501	3110	1555	2.61	25	230	200
5A	3501	3120	1560	0	0	230	240
1B	5244	2950	1475	16.24	100	238	250
2B	5250	3060	1530	12.25	50	248	200
3B	5240	3080	1540	8.16	75	248	200
4B	5253	3100	1550	3.99	25	230	300
5B	5257	3120	1560	0	0	220	320
1C	6995	2950	1475	21.44	100	240	475
2C	7006	3070	1535	16.23	75	230	420
3C	7004	3095	15475	10.8	50	245	370
4C	7004	3100	1550	5.41	25	230	290
5C	7001	3120	1560	0	0	230	250 PSI

Figure D-9 Qualification pump pressure pulsations per MIL-P-19692B

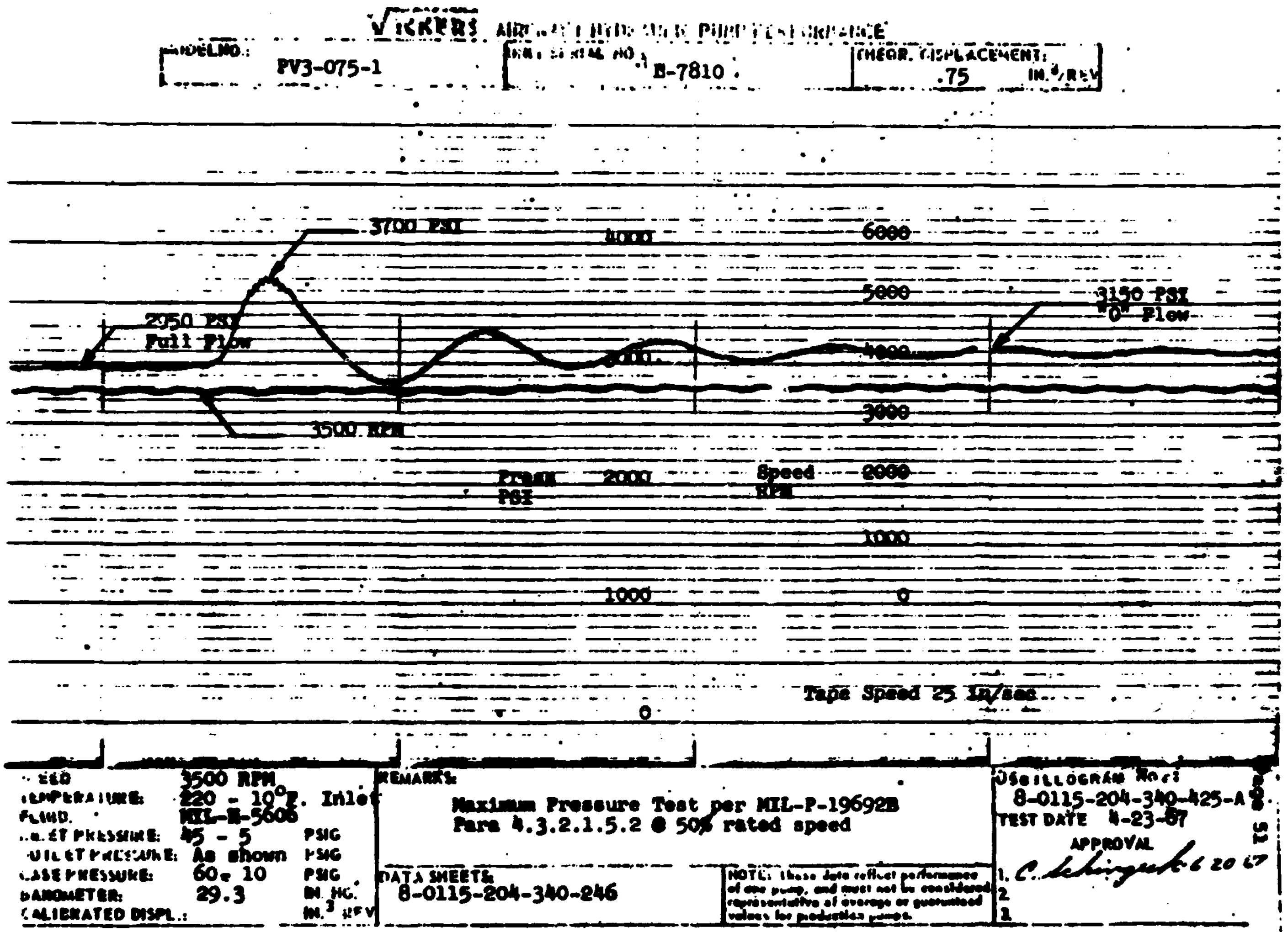


Figure D-10 Qualification pump transient pressure at 3500 rpm

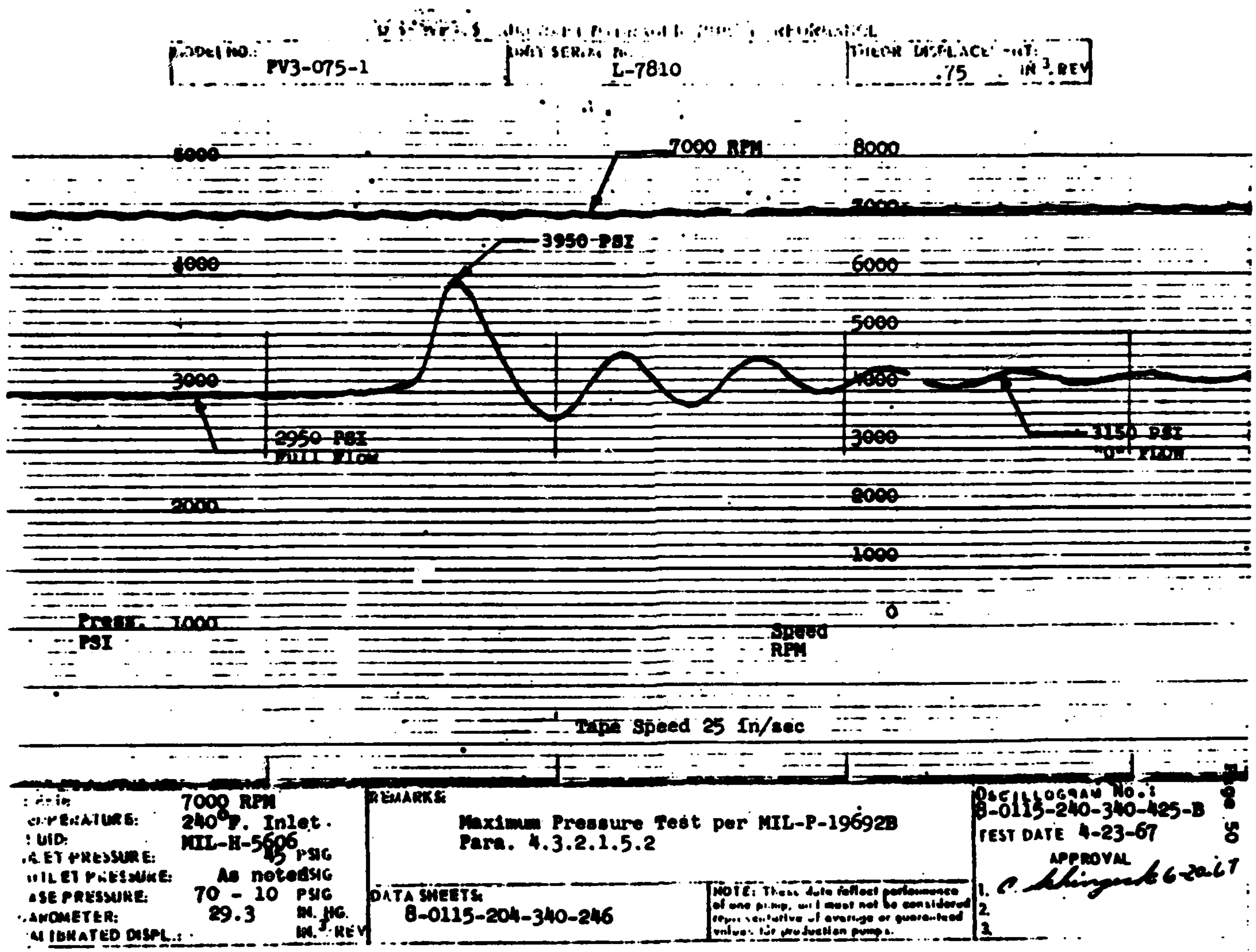


Figure D-11 Qualification pump transient pressure at 7000 rpm

APPENDIX E

SPERRY-VICKERS REPORT NO. 03-792030
TEARDOWN INSPECTION OF PV3-075-15 PUMP AFTER 50-HOUR TEST
IN AO-8 FLUID AT THE BOEING COMPANY
PROJECT NO. 8-1102-210 DATED JUNE 15, 1979

REPORT NO. 03-792030

SPERRY VICKERS

TROY, MICHIGAN 48084

REPORT
ON

Teardown Inspection of PV3-075-15
Pump After 50 Hour Test in AO-8
fluid at the Boeing Company

PROJECT NO. 8-1102-210

DATE RELEASED June 15, 1979

PREPARED BY N. D. Pedersen CHECKED BY

APPROVED BY K. F. Becker 6/15/79 CONCURRED BY

RELEASED BY

APPROVED FOR OUTSIDE RELEASE K. F. Becker 6-15-79

1960-B Rev. 1-77

SPERRY VICKERS

REPORT NO. 03-792030

PAGE 1

ABSTRACT

A standard PV3-075-15 in-line piston pump was fitted with special PNF seals at Boeing's Wichita plant and subjected to a 50-hour operating test in Halocarbon AO-8 fluid. At the termination of the test, the pump was disassembled and sent to Sperry Vickers for review. This report discusses the observed conditions of the pump, and makes recommendations based on the review.

TABLE OF CONTENTS

INTRODUCTION

The PV3-075-15 pump was disassembled after a 50-hour test in Halocarbon AO-8 fluid at Boeing's Wichita plant on April 12, 1979. The pump parts were drained of fluid and air dried. The parts were received at Sperry Vickers on April 16 and examined by a group of Sperry Vickers engineers on April 19. Visual examination without complete disassembly of such items as the ball thrust bearing from the drive shaft revealed a thin deposit of a dark substance on the bronze parts of the pump, sluggish rotation of the drive shaft, hot spots on the pistons, erosion of the piston shoes, and a black, gummy deposit in the housing adjacent to the shaft seal.

In general, preliminary examination of the pump showed it to be in fairly good condition.

More detailed examination, plus more complete disassembly revealed additional problems. The drive shaft in the area covered by the bearing inner race is deeply etched. The drive shaft ball bearing shows evidence of inadequate lubrication. The yoke pintle bearings also show evidence of inadequate lubrication and the yoke pintles showed evidence of rust. (Bearing condition statements herein are quotes from MPB Corporation--manufacturer of the bearings used in the pump.)

CONCLUSIONS & SUMMARY

1.0 All internal parts of the PV3-075-15 pump exhibit evidence of chemically attacked or etchings resulting in an apparent surface coating and/or discoloration. Reference attached 4 photographs of test hardware.

2.0 The test fluid appears to more severely attack surfaces of the pump where two surfaces are in close proximity such as shaft bearing inner race and the drive shaft.

3.0 The fluid has a degreasing property that leaves surfaces completely exposed to atmospheric corrosion.

4.0 The SEM-EDAX analysis were unable to distinguish or identify any elements other than the base material.

5.0 Two piston and shoe subassemblies and the carbon shaft seal element have been forwarded to Ed Snyder at WPAFB for additional investigations.

6.0 Evaluation of the mating ring deposit (black gum deposit) showed presence of hydrocarbon linkage and halogenated matter. No indication of viscosity improvers in the form of a poly-methocrylate could be detected.

7.0 Evaluation of the shaft seal carbon material was inconclusive due to the proprietary nature of the material by the manufacturers (Pure Carbon Co.)

8.0 The AO-8 fluid formulated for this test exhibits insufficient lubrication properties for rolling contact bearings.

9.0 Re-evaluation of the pump condition shows that continued operation on the same AO-8 fluid would result in premature failure.

RECOMMENDATION

1.0 Continuation of pump operation on the existing AO-8 fluid is not recommended.

2.0 Provisions should be considered on future pump evaluation to protect the pump materials from atmospheric corrosion following disassembly.

DESCRIPTION OF PUMP PARTS

1.0 General

Pump parts were discolored with the surface color ranging from a light yellow to a dark golden brown. In the majority of cases the surface color appears to be a thin film. However, in others the surface has an etched appearance.

2.0 Housings

The interior of the 355 T-6 aluminum housings have a distinct yellow color that is in some areas almost brown. The exterior of the mounting flange around the shaft seal location has a thick black gummy deposit.

3.0 Cylinder Block

The cylinder block has a dark golden brown color which has the appearance of being a very thin coating. On the face of the cylinder block, what appears to be a very thin powder can be removed with the finger nail.

4.0 Piston and Shoe Sub-assembly

Pistons are discolored but this may be a thin film of rust generated since the disassembly. In the central zone of the piston there is a hot spot or burn mark approximately 1/8 inch square. Shoes also exhibit the dark golden brown coating. Tests made by Sperry Vickers metallurgist are documented in the appendix. Two of the piston and shoe subassemblies have been sent to Ed Snyder at Wright Field for further tests.

4.0 Piston and Shoe Sub-assembly (Continued)

In addition, some of the shoes had light to moderate erosion in the lubricating slots radiating outward from the center hole.

5.0 Yoke and Shoe Wear Plate

Outer surface of yoke shows the yellow coating. Surface on which the shoe hold down plate is fastened has an etched pattern. Surface under wear plate on loaded side is also etched and the wear plate mating surface is etched in a matching pattern. Pintle bearings were removed with difficulty.

6.0 Drive Shaft

Drive shaft has yellowish coating. Surface of shaft under the inner race of the ball thrust bearing is heavily etched. There was a thick black substance present at this interface at disassembly.

7.0 Ball Thrust Bearing

The ball thrust bearing showed evidence of inadequate lubrication. Ball retainer which is made of bronze showed a similar dark brown coating on the cylinder block and piston shoes. The steel parts of the bearing were rusted but the probable time in which this occurred was after disassembly.

8.0 Pintle Bearings

These are full complement roller bearings. They also showed evidence of inadequate lubrication.

9.0 Shaft Seal Element

This part is manufactured as a resin impregnated carbon part. The resin is supposed to be inert. There is also a binder prior to sintering that the supplier considers to be proprietary. Tests of the black deposit were inconclusive in determining that either material contributed to the condition.

APPENDIX

1960
(REV. 1/77)

REPORT NO. 03- 792029

SPERRY VICKERS

TROY, MICHIGAN 48084

REPORT
ON

Metallurgical Evaluation
Fire Resistant Hydraulic
System Pump

PROJECT NO. 8-1102-210

DATE RELEASED June 15, 1979

PREPARED BY W. C. Scarborough CHECKED BY N. J. Pedersen

APPROVED BY Robert A Egge CONCURRED BY

RELEASED BY

APPROVED FOR OUTSIDE RELEASE K. F. Baker 6-15-79

INTRODUCTION

A PV3-075-15 pump (Mx-3 9687), was tested in a fire resistant hydraulic fluid (AO-8, Halocarbon Products Corp.) by the Boeing Co. Disassembly after test revealed discoloration and deposits on a number of the internal components.

PURPOSE

Identification of the discoloration on the bronze components and the tar like material on the mating ring.

CONCLUSIONS

(1) The discoloration on the bronze components was the result of chemical etching.

(2) Identification of the deposit on the mating ring was inconclusive. Only indications of halogen material and small amounts of hydrocarbon base matter were detected.

SUMMARY

1.0 Visual Examination

Discoloration in the form of apparent light rust was noted on the majority of the steel components. More serious indications of pitting were observed on selective components, namely the bearings and the back side of the drive shaft hub area.

All copper base materials exhibited an etched condition. No indication of build up was observed on this type of component. A black gummy deposit was present in the shaft seal area of the pump. Heaviest concentration of foreign material was deposited on the mating ring.

2.0 SEM Evaluation

Confirmed the etching reported in the visual examination. EDAX determinations failed to identify the source of attack.

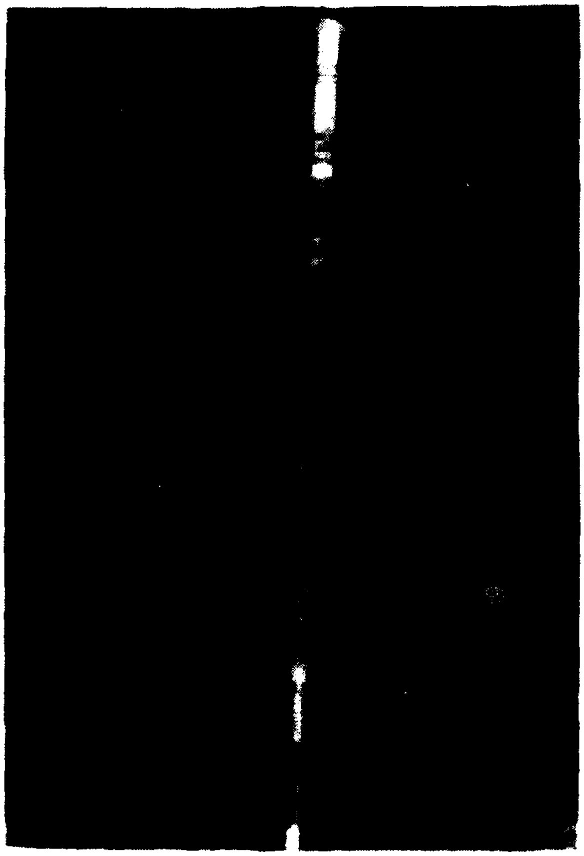
3.0 Infrared Spectrometer

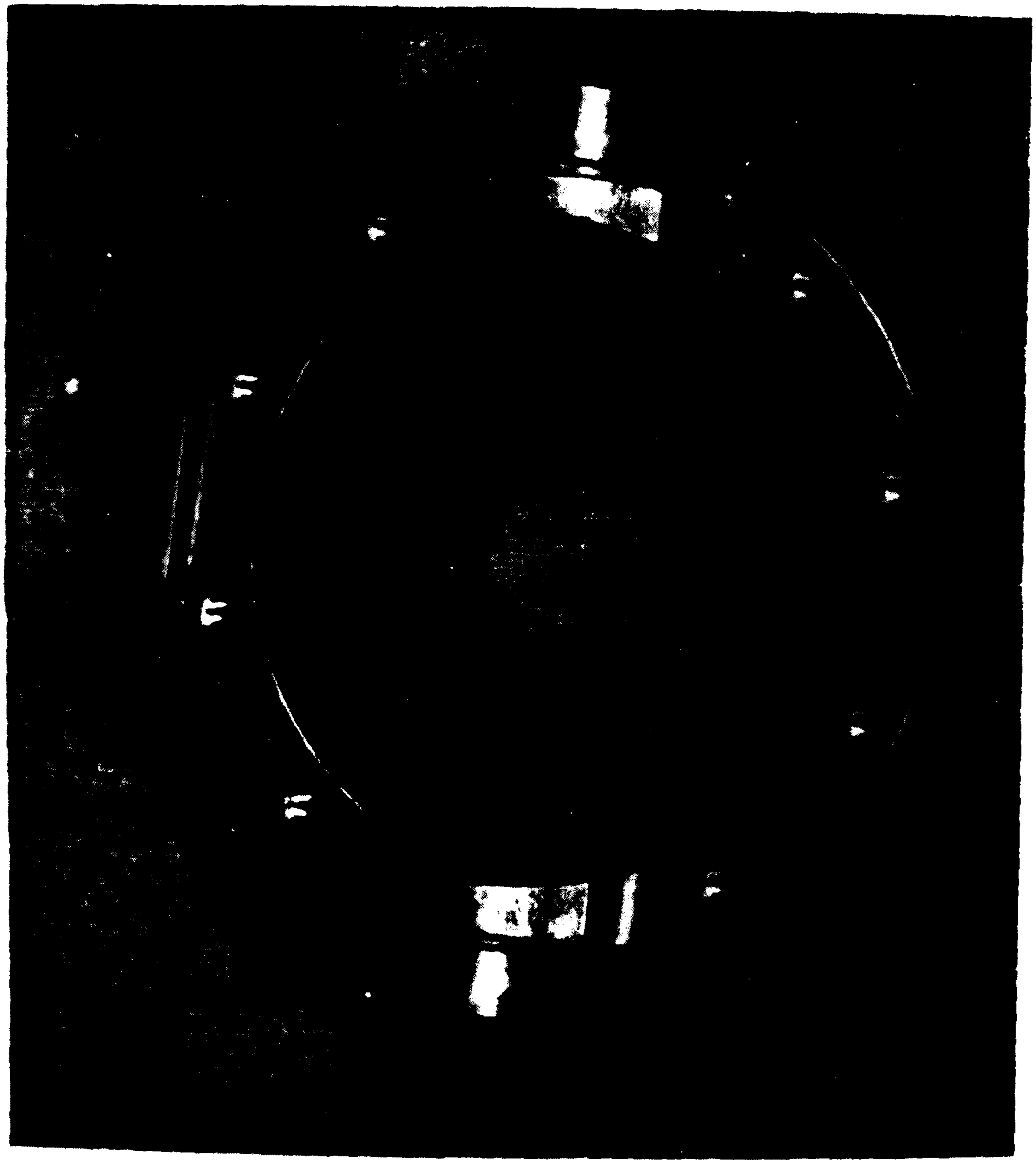
Spectrum scan for foreign material located on mating ring suggested the presence of hydrocarbon linkage and halogenated matter. No indication of viscosity improvers in the form of a poly-methacrylate could be detected.

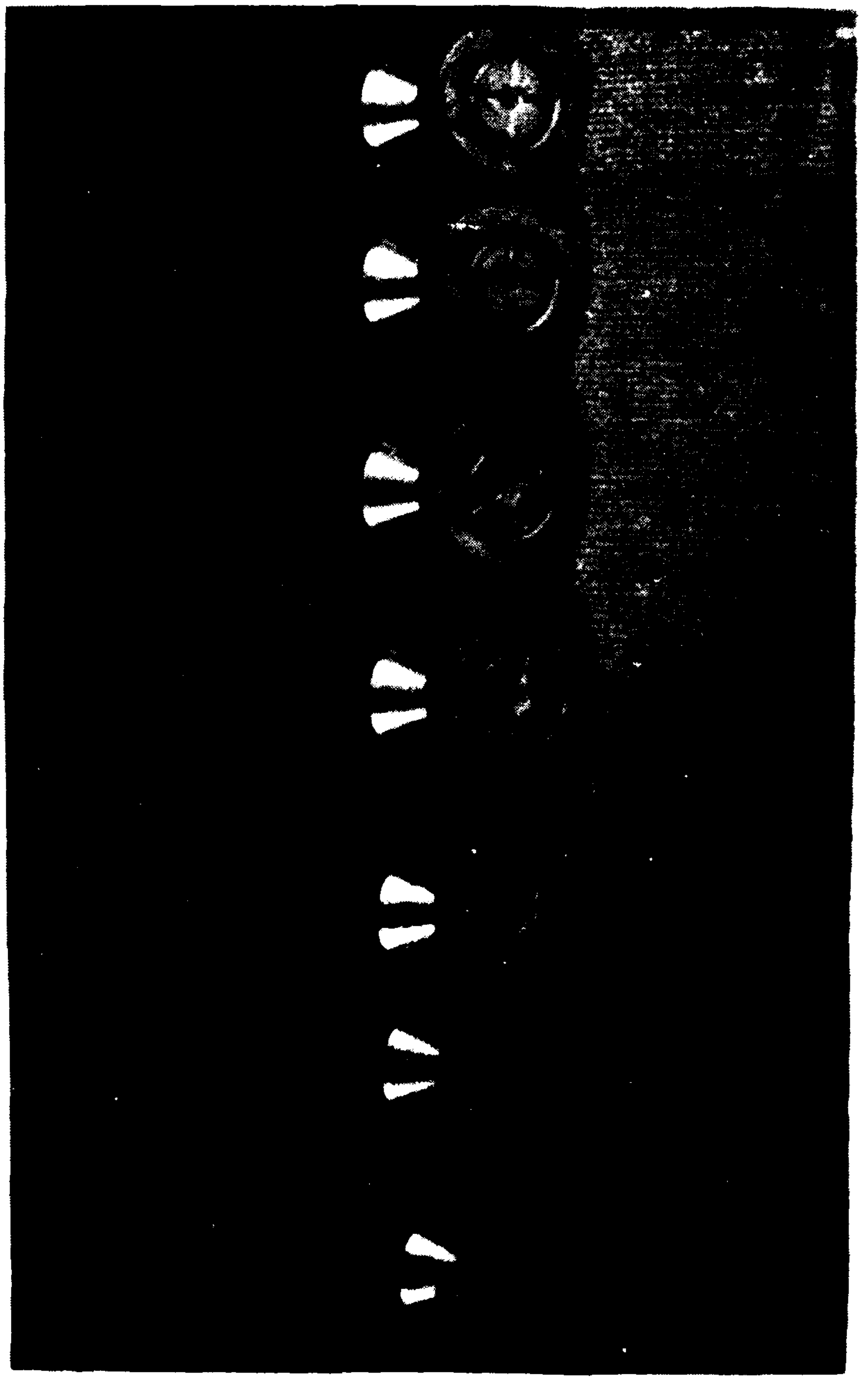
4.0 Fluid Evaluation

a) Moisture: Determinations by the Karl-Fischer method on fluid used in functional test yielded values of less than 10 parts per million.

b) Neutralization Number: No indication of acidity could be detected in the test fluid.







APPENDIX F

DISCUSSION OF THE UNIVERSITY OF DAYTON'S AUGER ANALYSIS OF THE PV3-075-15 HYDRAULIC PUMP AFTER BOEING 50-HOUR TEST

Discussion of Auger Analysis of Vickers Model No. PV3-075-15
Hydraulic Pump - After Boeing 50 hour test

The following parts from a Vickers Model No. PV3-075-15 pump following 50 hours of testing with unformulated Halocarbon A08 were analyzed using Auger spectroscopy: (1) retaining plate, (2) piston barrel, (3) piston shoe, (4) thrust bearing - inner race, (5) outer race, (6) bearing cage, and (7) yoke pintle bearing.

Attention was paid to bronze parts which had a high degree of darkening and to steel or bronze parts with a high degree of wear. Auger spectra were taken both at the surface (after acetone washing in an ultrasonic cleaner) and after ion-etching to a deeper level to remove surface adsorbed materials. All of the components had a high level of carbon on the surface which is attributed to residual oil still adhering to the parts. This carbon was not deep (absent on the ion etching scan) indicating that the carbon had not reacted with the metal. As illustrated by the data table, the darkening on the bronze was apparently due to a chemical reaction between the chlorine in the fluid and the copper in the bronze. The retaining plate and the cage appeared to be in stoiciometric ratios of either $CuCl_2$ or CuCl. On the retaining plate this layer was thicker, still present after ion-etching.

The steel components did not show evidence of reaction with the chlorine: only slight surface absorption of the fluid indicated by chlorine and carbon. The bearing and races do show small amounts of copper and chlorine together on the surface indicating some transfer of copper chlorine from the bronze cage or transfer of bronze to the steel surfaces and subsequent reaction with the fluid or its degradation products.

To summarize, Halocarbon A08 oil does form a dark reaction product layer with the copper in bronze but does not appear to react with the steel pump components.

C. E. Snyder

C.E. SNYDER
Fluids, Lubricants and Elastomers Branch
Nonmetallic Materials Division

AUGER ANALYSIS OF VICKERS MODEL NO. PV3-075-15
HYDRAULIC PUMP - AFTER BOEING 50 HR TEST

Approx. Atom % Composition

Part	Material	Cu	O	C	Cl	S	Zn	Sn	N	Na	Fe	Mn
Retaining Plate -surface	Bronze	13	4	47	26	4	<0.5	<0.5	<1	<2	<1	<1
Retaining Plate -after 35 min. ion-etching		37	4	13	35	2	<0.5	1	<1	<2	3	<1
Piston barrel -surface	Steel	<0.5	16	64	2	1	<0.5	<0.5	1	8	6	<1
Piston barrel -after 35 min. ion-etching		<0.5	13	5	<0.5	<0.5	<0.5	<0.5	<1	<2	76	<1
Shoe face -surface	Bronze	10	5	70	6	1	3	<0.5	1	<2	<1	<1
Shoe face -after 20 min. ion-etching		60	12	3	1	1	15	<0.5	<1	<2	<1	3
Bearing, inner race-surface	Steel	2	19	43	6	4	<1	<0.5	4	7	13	<1
Bearing, inner race-after ion-etching ~500 Å		<1	30	8	2	<1	<1	<0.5	<1	<2	53	<2
Bearing, outer race-surface	Steel	2	15	60	6	2	<1	<0.5	2	3	9	<1
Bearing, outer race-after ion-etching ~500 Å		1	42	4	1	1	<1	<0.5	<1	<3	43	<2

(continued)
AUGER ANALYSIS OF VICKERS MODEL NO. PV3-075-15
HYDRAULIC PUMP - AFTER BOEING 50 HR TEST

Approx. Atom % Composition

Part	Material	Cu	O	C	Cl	S	Zn	Sn	N	Na	Fe	Mn
Cage-surface	Bronze	13	5	46	24	4	<1	1	3	3	1	<1
Cage-after ion-etching ∼4000Å		68	8	6	3	3	<1	3	<1	<3	<2	<2
Yoke pintle bearing -surface	Steel	<1	5	82	2	2	<1	<0.5	2	<2	2	<2
Yoke pintle bearing -after ion-etching ∼500Å		<1	24	46	1	<1	<1	<0.5	2	<2	20	<1

APPENDIX G

DISCUSSION OF THE MATERIALS LABORATORY'S PUMP MATERIALS
AND HYDRAULIC FLUID ANALYSIS
AFTER BOEING 50-HOUR TEST

DEPARTMENT OF THE AIR FORCE
AIR FORCE MATERIALS LABORATORY (AFSC)
WRIGHT-PATTERSON AIR FORCE BASE, OHIO 45433

REPLY TO ATTN OF: MBT (C.E. Snyder, Jr.) 21 May 1979

SUBJECT: Characterization of Samples from 50 Hour Pump Test Performed by Boeing Aircraft Co. under Contract F33615-76-C-2064

TO: AFAPL/POP-2 (Bruce Campbell)

1. Characterizations of the used fluid samples from subject pump test and the two black tarry samples taken from the coupling shaft spline have been completed.

2. The two black tarry samples were taken at the end of 41 hours and at the completion of the 50 hour pump test, respectively. Analysis of the first sample by infrared spectroscopy revealed no evidence of the A08 fluid present in the hard, tacky material. The only non-carbon constituent appeared to be the binder of the carbon face seal. The second sample which was taken at the end of the 50 hour test period was quite liquid and was found to contain A08 fluid by infrared spectroscopy. In both cases, when the samples were rinsed from the infrared plates, a drop of chloroform used as a cleaning solvent readily dissolved the non-carbon component, leaving behind a fine black particulate residue which was completely insoluble in any solvent. This was assumed to be carbon because of its appearance and behavior.

3. The fluid sample taken at 41 hours and 50 hours of the pump test were assigned sample numbers MLO79-110 and MLO-79-111, respectively. The unstressed fluid sample used for this pump test was MLO78-299. The data on all three fluid samples are shown in the attached Table I. Analysis of the data indicate that only minor changes in physical properties of the fluid occurred. That coupled with the fact that no unsaturation was found which would be present if degradation of the fluid occurred indicates that the fluid underwent insignificant degradation during the 50 hour pump test. Spectroscopic oil analysis of these samples revealed that all metals were at 0 ppm. This means that any wear materials generated during the test were not dispersed or dissolved in the fluid samples.

4. In order to better understand the reason for the reddish deposit on the bronze parts of the pump, a corrosion oxidation test using all copper metal test specimens was conducted for 15 days (360 hours) at 275°F. The data on fluid

property and metal weight change are shown for duplicate tests on the attached Table II. In one tube, the fluid viscosity change and acid number change were insignificant and the metal weight changes were all acceptable and the metals did not show much resemblance to the reddish color of the brass pump parts. The duplicate test, however had a rather significant increase in acid number and the copper test panels did have a reddish color deposit on the surface that could be removed by scratching with one's fingernail. As can be seen from the very small weight changes in these metal specimens the reaction occurring is very mild and it is not known how deterimental it would be to pump performance.

5. If there are any questions, please contact the undersigned.

Carl E. Snyder

CARL E. SNYDER
Fluids, Lubricants and Elastomers Branch
Nonmetallic Materials Division

2 Atchs
1. Table I
2. Table II

2

ATCH 1

Table I - Fluid Property Data on Boeing Pump Test Samples Versus Unstressed Halocarbon A08

	MLO78-299	MLO79-110	MLO79-111
Viscosity at 100°F, cs	7.58	7.38	7.31
% Change		-2.6	-3.6
Viscosity at 210°F, cs	2.21	2.19	2.25
% Change		-0.9	+1.8
Gas Chromatography		No change	No change
Differential Infrared		Very small peaks at 2960, 2930 cm^{-1}	Very small peaks at 2960, 2930 cm^{-1}
Acid Number, mg KOH/g	0.03	0.08	0.10
Test for Unsaturation	Saturated	Saturated	Saturated
Particulate Contamination, mg/100ml		2.03	2.18
Spectroscopic Oil Analysis for Metals		Negative	Negative

3

ATCH 2

Table II - Corrosion Oxidation Test of ML078-299 using Copper Metal Specimens - 275°F - 15 Days

	Test I	Test II
Viscosity Change at 100°F, %	-1.2	-2.6
Acid Number Change, mg KOH/g	0.04	0.43
Gas Chromatography	No change	No change
Differential Infrared Spectroscopy	No change	No change
Test for Unsaturation	Saturated	Saturated
Metal Weight Change, mg/cm^2		
Copper I	-0.14	+0.09
II	-0.12	+0.07
III	-0.10	+0.09
IV	-0.12	+0.10
V	-0.13	+0.11

APPENDIX H
LONG-TERM PUMP TEST PLAN

The long-term pump test outlined herein will meet the requirements specified in Section F, paragraph 4.3.5 of contract F33615-76-C-2064. The selected pump is a Sperry-Vickers PV3-075-15 model with rated speed of 7,000 rpm and a rated inlet temperature of 240F. The system hydraulic fluid will be Halocarbon AO-8 with an anti-wear additive. The pump operating characteristics and probable life expectancy will be determined with the AO-8 fluid. Pump teardown inspections will be accomplished to determine any design deficiencies. Testing will be halted for any sudden performance degradation and a teardown inspection performed.

All testing will be performed in a test setup as shown in Figure H1. Unless otherwise specified, the reservoir pressure will be adjusted to produce 100 ± 5 psia at the pump inlet for all steady-state conditions and for the highest flow condition during all tests where flow is suddenly changed or cycled. The shaft seal leakage will be monitored and recorded throughout the testing. Instrumentation will be state-of-the-art and recently calibrated.

<u>Sequence of Inspections and Tests</u>

Initial Inspection and Assembly
Rotation Torque Check
Proof Pressure Test
Initial Break-in Run
Rotation Torque Check
Teardown, Inspection and Reassembly
Rotation Torque Check
Proof Pressure Test
Abbreviated Run-in
Fifty-Hour Run
Rotation Torque Check
Teardown, Inspection and Reassembly
Rotation Torque Check

- Proof Pressure Test
- Abbreviated Run-In
- Calibration Test
- Maximum Transient Pressure Test
- Response Time Test
- Heat Rejection Test
- Low Temperature Test
- Endurance Test
    - Calibration Test at 7,000 rpm
    - Start-stop Cycles
        - Full-load Cycles
        - No-flow Cycles
    - Normal Endurance Test
    - Calibration Test at 7,000 rpm
    - Start-stop Cycles
        - Full-load Cycles
        - No-flow Cycles
    - Thermal Cycles
    - Calibration Test at 7,000 rpm
- Rotation Torque Check
- Teardown and Inspection

Inspections

During the initial inspection and assembly, measurements of piston-shoe assembly clearance, cylinder-block piston bores, piston diameters, bronze shaft seal nose details, and any other dimensions deemed necessary will be taken. During subsequent teardowns and inspections, the piston-shoe assembly clearance will be measured; and, any parts showing significant or unusual wear will be measured and photographed. In addition, at the end of testing, the test system filters will be inspected for pump wear particles and fluid breakdown products.

Test Procedures

Rotation Torque Check

The torque required to rotate the drive shaft by hand will be measured with all fluid ports open. Values greater than 30 lb-in for breakaway and 25 lb-in for running will be cause for concern and investigation.

Proof Pressure Test

With the unit at a standstill, and the inlet port and pressure discharge port plugged, 500 psig will be applied to the case drain port for five minutes and any external leakage measured. The pressure at the case drain port will then be reduced and any external leakage measured while holding one foot head of fluid at the case drain port for 30 minutes. If external leakage exceeds the following limits by any appreciable amount; the pump will be disassembled, the leaking seal(s) inspected and reworked or replaced, and the test rerun:

Shaft seal static leakage: one drop in 30 minutes.
Other static seal leakage: slight wetting insufficient to form a drop

Initial Break-in Run

The pump will be started and run for 5 minutes at 3,500 rpm with 1,500 psi discharge pressure. Discharge pressure will then be gradually increased to 2,850 psi at full flow (10.2 gpm minimum) (by adjusting the compensator if necessary) and held at that setting for 25 minutes. Pump speed will then be increased to 7,000 rpm and the pump run for 30 minutes while maintaining 2,850 psi discharge pressure and full flow (20.5 gpm minimum). During these runs, inlet fluid temperature may range from ambient to 180F. With the valves in the discharge line closed, the pump should produce a discharge pressure of 3,025 $\pm$25 psig. If it is not, the compensator will be adjusted to bring it within this range. Then, the pump speed will be increased to 7,700 rpm and the inlet fluid temperature stabilized at 180F for the following pressure drift and stability checks:

a. With the discharge flow at 5 gpm, the discharge line will be suddenly closed and the discharge pressure measured. It should be 3,000 ±50 psig and no oscillations (other than pressure pulsations) should occur for more than one second.

b. At the same operating condition, with the discharge line closed, the discharge pressure will be observed during 5 minutes of continuous running. The mean value should not drift more than 25 psi above or below the inital cutoff pressure.

c. The discharge flow will be increased to 1.0 ±0.5 gpm and the pump speed slowly decreased to 3,500 rpm. Discharge pressure will be observed for any oscillations persisting for more than one second.

d. With the discharge flow increased to 3 gpm, the discharge line will be suddenly closed and the discharge pressure measured. It should be 3,000 ±50 psig and no oscillations should occur for more than one second.

e. At the same operating condition, with the discharge line closed, the discharge pressure will be observed during 5 minutes of continuous running. The mean value should not drift more than 25 psi above or below the inital cutoff pressure.

The pump speed will be increased again to 7,700 rpm and the inlet fluid temperature stabilized at 240F. The foregoing pressure drift and stability checks will be repeated.

Abbreviated Run-In

The pump will be started and run for 5 minutes at 3,500 rpm and 1,500 psi discharge pressure. Discharge pressure will then be gradually increased to 2,850 psi at full flow (10.2 gpm minimum) (by adjusting the compensator if necessary) and held at that setting for 25 minutes. Pump speed will then be increased to 7,000 rpm and the pump run for 30 minutes while maintaining 2,850 psi discharge pressure and full flow (20.5 gpm minimum). During these runs, inlet fluid temperature may range from ambient to 180F. With the valves in

the discharge line closed, the pump should produce a discharge pressure of 3,025 ±25 psig. If it is not, the compensator will be adjusted to bring it within this range.

Fifty-Hour Run

The pump will be run at 7,000 rpm with the inlet fluid temperature at 240F and with the discharge flow and pressure cycled repetitively for 50 hours from one of the following conditions to the other:

5 sec. at full flow (20.5 gpm min.) at 2,700 psig min.
5 sec. at zero flow at 3,000 ±50 psig

(These requirements are from MIL-P-19692C, Phase 7 of Table IV Endurance Test Conditions (Normal Test).)

Calibration Test

A calibration test will be conducted per the requirements of MIL-P-19692C, paragraph 4.3.2.1.4 with the inlet fluid temperature at 180F, except that the test conditions of minimum operating speed and 25 percent of rated speed will be deleted.

Maximum Transient Pressure Test

This test will be conducted per the requirements of MIL-P-19692C, paragraphs 4.3.2.1.5 and 4.3.2.1.5.2, with the inlet fluid temperature at 240F. Four runs will be made as follows:

1. At 3,500 rpm with the basic test system fluid compliance volume (approx. 50 cu.in.).

2. At 3,500 rpm with an additional 200 cu.in. fluid compliance volume.

3. At 7,000 rpm with the basic compliance volume.

4. At 7,000 rpm with an additional 200 cu.in. volume.

Response Time Test

This test will be conducted per the requirements of MIL-P-19692C, paragraphs 4.3.2.1.5 and 4.3.2.1.5.3, with the inlet fluid temperature at 240F. Two runs will be made as follows, all with the additional 200 cu.in. fluid compliance volume open to the test circuit, at each of three speeds (3500, 5250, and 7000 rpm) for a total of six runs:

1. With full discharge flow at 2,850 psi suddenly shut off to a steady-state zero-flow condition at 3,000 psi.

2. With discharge flow suddenly increased from zero flow (at 3,000 psi) to full flow at 2,850 psi steady-state pressure.

Heat Rejection Test

the pump heat rejection rate will be determined per the requirements of MIL-P-19692C, paragraphs 4.3.2.1.6 and 4.3.2.1.6.1.

Low Temperature Test

The pump will be tested from -65 ±5F per the requirements of MIL-P-196692C, paragraph 4.3.2.1.8.

Endurance Test

The pump will be tested per the requirements of the following paragraphs from MIL-P-19692C in the following sequence:

1. Calibration Test at 7,000 rpm (Para. 4.3.2.1.4)

   This test will be run at 7,000 rpm with the inlet fluid temperature at 180F.

2. Initial Start-Stop Cycles (Para. 4.3.2.1.9.4)

All start-stop cycles will be run with the added 200 cu.in. compliance volume open to the test circuit. Temperatures will range from ambient to rated and will be recorded.

a. Initial Full-Load Cycles (Para. 4.3.2.1.9.4.1)
A total of 125 full-load start-stop cycles will be run prior to the normal endurance test with the load orifice adjusted to 2700 psi at full flow.

b. Initial No-Flow Cycles (Para. 4.3.2.1.9.4.2)
A total of 16 no-flow start-stop cycles will be run prior to the normal endurance test.

3. Normal Endurance Test

A total of 700 hours, consisting of the six phases specified in Table H1 herein, will be run in the order listed.

4. Calibration Test at 7,000 rpm (Para. 4.3.2.1.4)

This test will be run at 7,000 rpm with the inlet fluid temperature at 180F.

5. Final Start-Stop Cycles (Para. 4.3.2.1.9.4)

All start-stop cycles will be run with the added 200 cu.in. compliance volume open to the test circuit. Temperatures will range from ambient to rated and will be recorded.

a. Final Full-Load Cycles (Para. 4.3.2.1.9.4.1)
A total of 125 full-load start-stop cycles will be run following the endurance test with the load orifice adjusted to 2700 psi at full flow.

b. Final No-Flow Cycles (Para. 4.3.2.1.9.4.2)
A total of 16 no-flow start-stop cycles will be run.

6. <u>Thermal Cycles (Para. 4.3.2.1.9.7)</u>

The pump compensator will be readjusted to produce a discharge pressure of 3,025 ±25 psig at zero flow, and the inlet pressure readjusted to 100 ±5 psia.

Eight thermal cycles will be run with the pump started at -20F and operated at 2,850 psi discharge pressure, with the speed brought immediately up to 3,500 rpm and gradually increased to 7,000 rpm, until the inlet fluid temperature reaches 240F.

7. <u>Calibration Test at 7,000 rpm (Para. 4.3.2.1.4)</u>

The pump compensator setting to produce a discharge pressure of 3,025 ±25 psig at zero flow and the 100 psi inlet pressure will be maintained. The test will be run at 7,000 rpm with the inlet fluid temperature at 180F. The results of the three endurance test calibrations will be plotted on one chart to show the effect of the endurance test on pump performance.

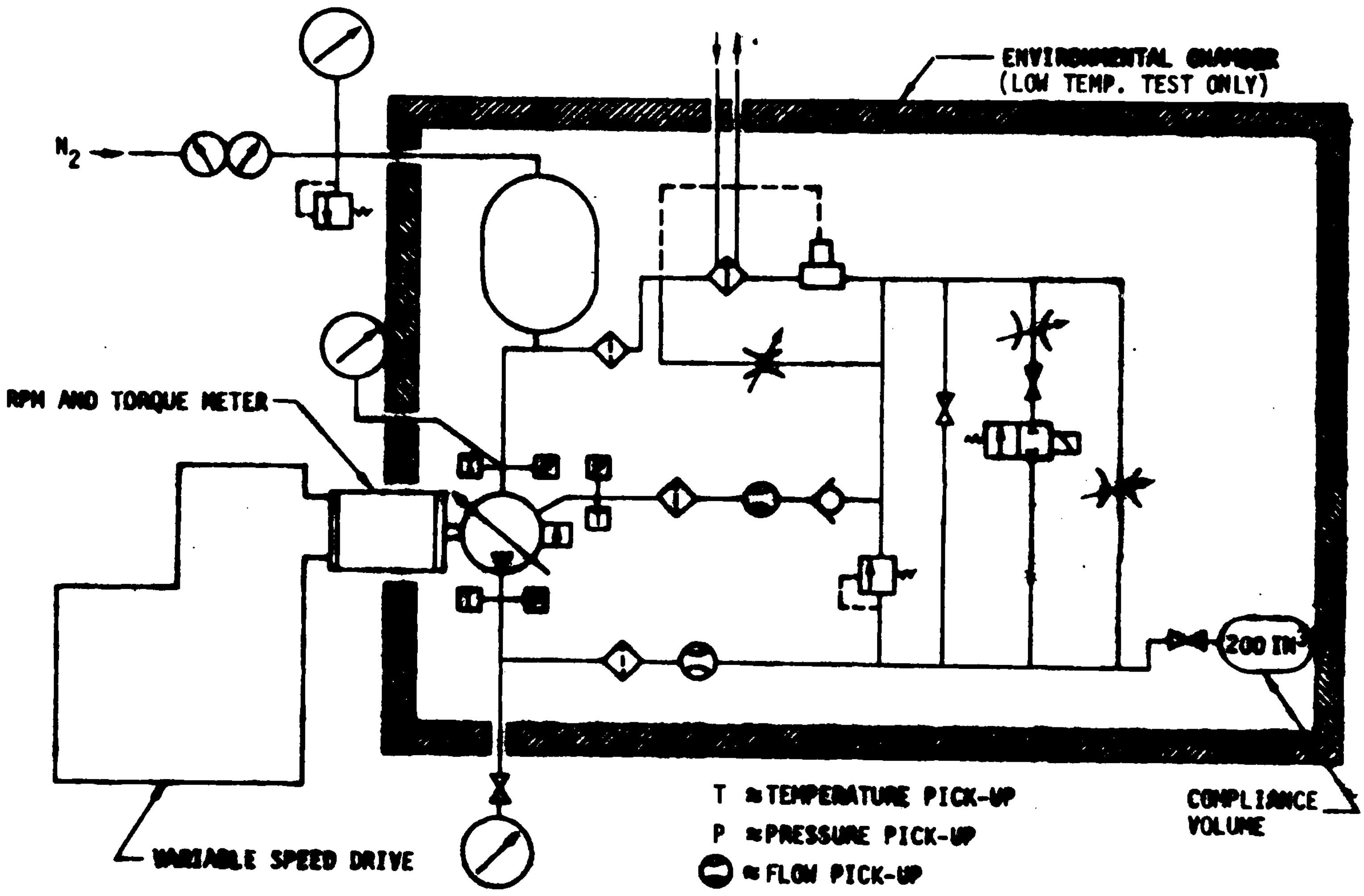


Figure H1. Long-term pump test setup

TABLE H.1 PUMP ENDURANCE TEST CONDITIONS

PHASE	SPEED rpm	DURATION hr	INLET FLUID TEMP. deg. F	CYCLES					
				FLOW gpm	PRESSURE psi	DURATION min	FLOW gpm *	PRESSURE psi	DURATION min
1a	3500	150	170	0	3000	9	10.5	2750	1
1b		50	215						
2a	5200	150	170	0	3000	9	15.75	2750	1
2b		50	215						
3a	6000	75	170	0	3000	9	18.0	2750	1
3b		25	215						
4a	6500	75	170	0	3000	9	19.5	2750	1
4b		25	215						
5a	7000	37.5	170	0	3000	9	21.0	2750	1
5b		12.5	215						
6a	7700	37.5	170	0	3000	9	23.1	2750	1
6b		12.5	215						

*The flow rates shown in Column 8 are the approximate full-flow values for the speeds specified.

APPENDIX I

SPERRY-VICKERS REPORT NO. 03-802024
TEARDOWN INSPECTION OF PV3-075-19 PUMP
AFTER 656-HOUR ENDURANCE TEST IN AO-8 FLUID
AT THE BOEING COMPANY
PROJECT NO. 8-1102-210 DATED APRIL 10, 1980

1980
(REV. 1/77)

REPORT NO. 03- 802024

SPERRY VICKERS

Jackson, MS 39206

REPORT
ON

~~TEARDOWN~~ INSPECTION OF PV3-075-19
PUMP AFTER 656 HOUR ENDURANCE
TEST IN AO-8 FLUID AT THE BOEING COMPANY

PROJECT NO. 8-1102-210

DATE RELEASED April 10, 1980

PREPARED BY N. F. Pedersen CHECKED BY

N. F. Pedersen

APPROVED BY F. W. Perian CONCURRED BY

RELEASED BY

APPROVED FOR OUTSIDE RELEASE

ABSTRACT

The PV3-075-19 pump is an inline piston, positive displacement, variable delivery pump. This pump is identical to a PV3-075-15 pump except it has a bronze shaft seal element and PNF seals for operation in A0-8 fluid. The PV3-075-19 was run for 57:25 hours of operation at the Boeing Wichita test facility when a valve block/ cylinder block smear interrupted the test. The pump was refurbished with a new valve block, a new yoke actuator piston and a new number 7 pumping piston. The pump was then run 656:24 hours when it failed again while operating at 7000 RPM. This has led to the conclusion that development effort is required for inline pumps to operate with A0-8 fluid and provide life and speed capability equal to pumps that operate with MIL-H-5606 fluid.

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INTRODUCTION

The PV3-075-19 pump is an inline piston, positive displacement, variable delivery, pump. This pump is identical to a PV3-075-15 pump except it has a bronze shaft seal element and PNF seals for operation in A0-8 fluid. When building the PV3-075-19 pump, precautions were taken to insure the parts did not contact any other hydraulic fluid or operate in any other hydraulic fluid other than A0-8 fluid prior to operation in the customer's A0-8 fluid test circuit. An A0-8 fluid test circuit was not available at Sperry Vickers - Jackson Plant.

The PV3-075-15 is sold to General Dynamics for the F-16. The F-16 Hydraulic System operates with MIL-H-5606 fluid. The PV3-075-15 pump was selected for the initial A0-8 fluid test because it was in production and is the desired displacement. The initial test in the A0-8 Program consisted of replacing the seals of a PV3-075-15 with PNF seals at Boeing Wichita. The pump was then run through a 50 hour test that included overspeed and overpressure. The results of the inspection of this pump was reported in Technical Report 03-792030 dated June 15, 1979. At that time it was recommended that a new pump (PV3-075-19) be released for the A0-8 fluid tests. The pump would include a bronze element, PNF seals, and would not be operated (run in) on any fluid but A0-8. It was proposed that this pump would then be run in a 750 hour test with an improved A0-8 fluid. The test would not include overpressure or overspeed

runs. After 57:25 hours of operation at the Boeing Wichita facility a valve block/cylinder block smear interrupted the test. The pump was refurbished with a new valve block, a new yoke actuator piston and a new number 7 pumping piston. The pump was then run an additional 656:24 hours until it failed again when operating at 7000 RPM.

The parts of the PV3-075-19 pump were in better condition after 650 hours of operation than the parts of the PV3-075-15 were after the 50 hour test. This may be attributed to the improvers added to the A0-8 fluid. The failure of the cylinder block/valve block interface that occurred at 57 hours and again after an additional 656:24 hours leads to the recommendation for development effort for pumps operating in A0-8 fluid systems.

INSPECTION OF PUMP PARTS AND COMMENTS

1. Cylinder Block - Valve Block Interface

The bronze face of the cylinder block was badly scored with a great deal of material transferred to the valve block surface. The valve block was cracked at the drain holes leading from the bearing cavity. These cracks indicate heating of the surface to a temperature above the transformation point.

This failure is typical of one due to overloading an oil film bearing.

The load of the oil film bearing can be changed by changing the pressure balance to reduce the load and/or increasing the Kingsbury area. Development effort is indicated in this area.

2. Yoke Actuator Piston

Approximately 3/4" of the lower portion was severely galled. This apparently had little effect on the stability of the pump since it probably occurred early in the pump operation. Galling had already begun during the first 50 hour test and this part was replaced at the teardown. The mating bore in the aluminum housing was not galled. This piston is 52100 bearing steel. One of a higher hot hardness tool steel would probably improve the wear.

3. Pistons and Shoes

The surface of the pistons are discolored as they were at the 50 hour inspection but with no apparent further deterioration. This discoloration is apparently a build-up of material since the piston to bore clearance has been decreased by .0003 in every case. The shoes are eroded at the inside diameter at the end of the grooves in the pocket pads but to not much greater extent than at the 50 hour inspection.

Piston and shoe end play is .001 to .0025 with the maximum play in piston No. 8 which was formerly .0005.

4. The splines on the drive shaft at the cylinder block attachment are worn more than would be expected with MIL-H-5606. The wear was not enough to cause a performance problem after 713:49 hours of operation. A tool steel drive shaft would probably improve the wear characteristics.

The etching beneath the thrust bearing that had occurred on the first pump was virtually non-existent after this 700 hour test.

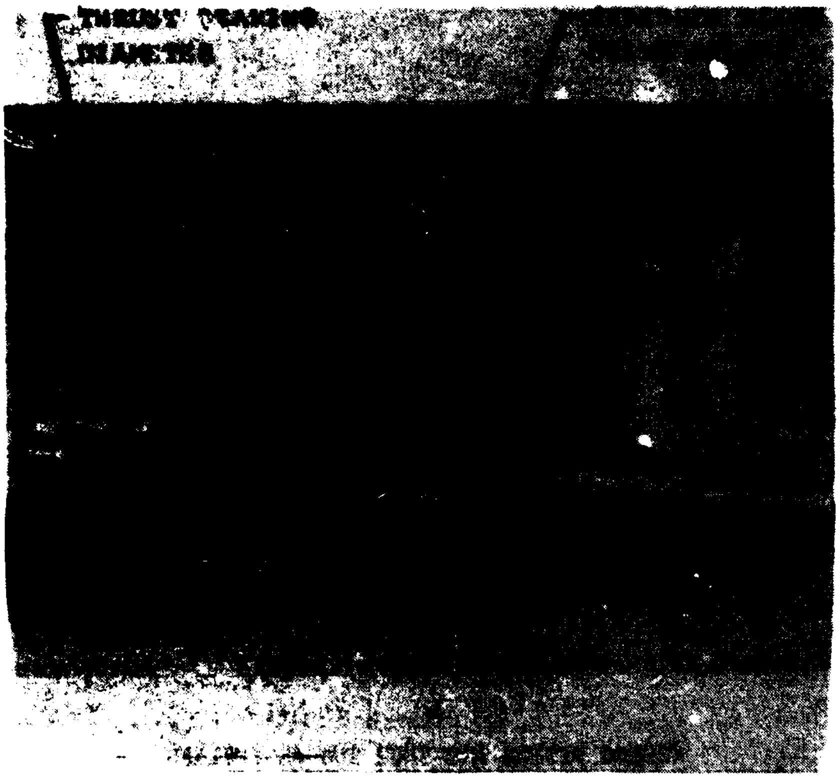
5. <u>Seals</u>

All static seals were softened and most were leaking a black tarry substance to the outside of the pump. An improved seal compound would be required for increased pump life.

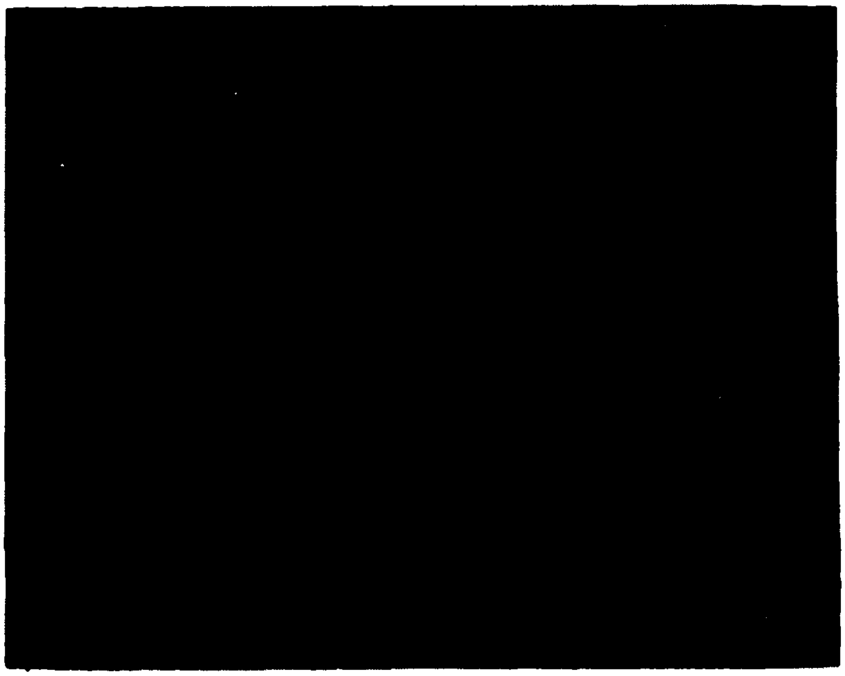
CONCLUSIONS

The following conclusions are based on the limited testing with A0-8 fluid at Boeing Wichita, examination of hardware after 713:24 hours of operation and the test data provided by Boeing.

Existing inline pumps require development for operation in A0-8 fluid. Material and design changes are required to equal the rated speeds and the life of pumps operating in MIL-H-5606 fluid systems.



317090 DRIVE SHAFT AFTER 713:49
HOURS OPERATION IN AO-8 FLUID

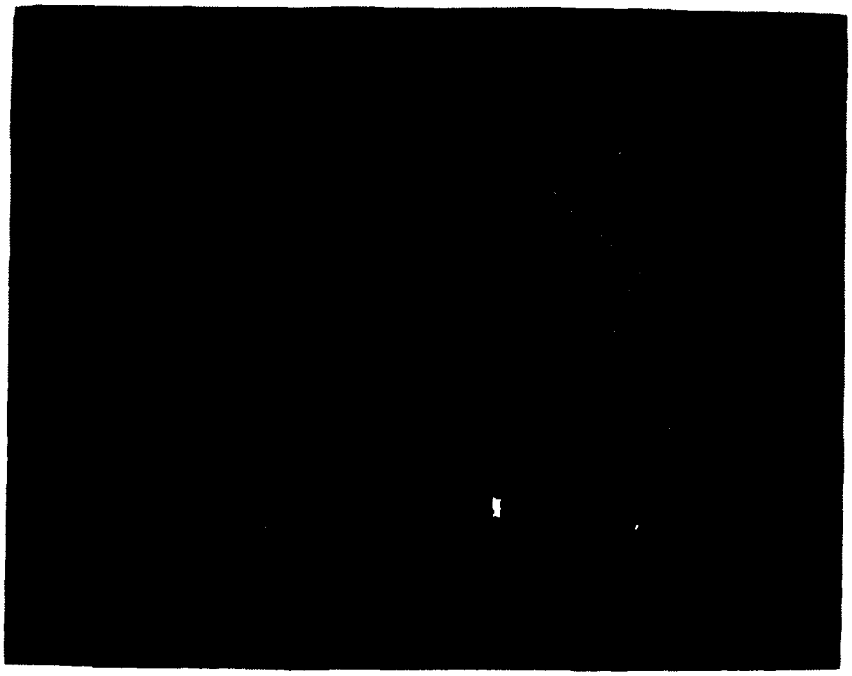


428200 PISTON AND SHOE SUB-ASSEMBLIES AFTER 713:49 HOURS OPERATION IN AO-8 FLUID (NOTE-NUMBER 7 PISTON ACCUMULATED 656:24 HOURS; THE ORIGINAL #7 PISTON WAS REPLACED AFTER THE FIRST 57:25 HOUR RUN)

317078 SHOE RETAINER PLATE
AFTER 713:49 HOURS OPERATION
IN AO-8 FLUID

428200 PISTON AND SHOE SUB-ASSEMBLY AFTER 713:49 HOURS OPERATION IN AO-8 FLUID

345343 PISTON SHOE BEARING
PLATE AFTER 713:49 HOURS OF
OPERATION IN AO-8 FLUID



326719 HOLD DOWN PLATE
RETAINER AFTER 713:49
HOURS OF OPERATION IN
AO-8 FLUID

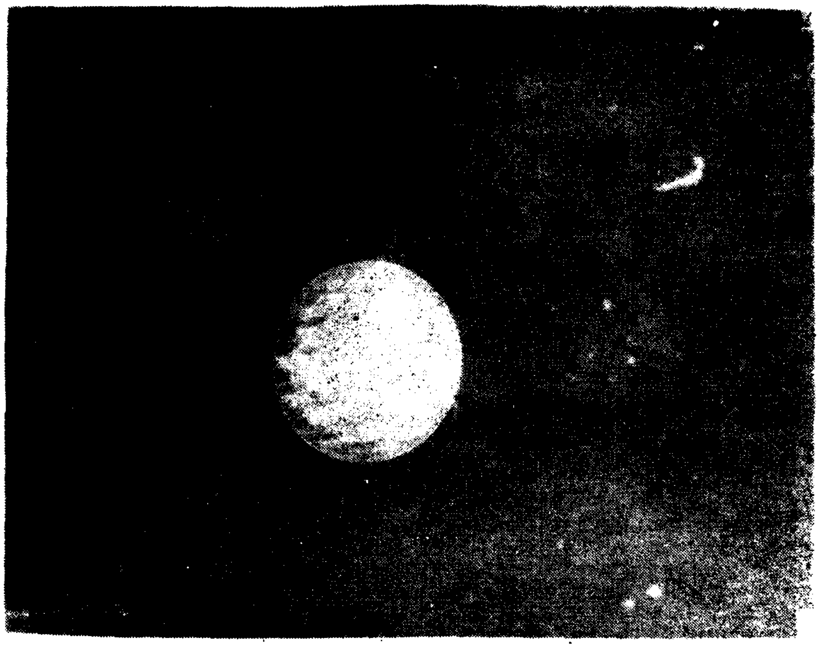
317084 ACTUATOR PISTON AFTER
656.24 HOURS OF OPERATION IN
AO-8 FLUID

KINGSBURY AREA

CYLINDER BLOCK FACE

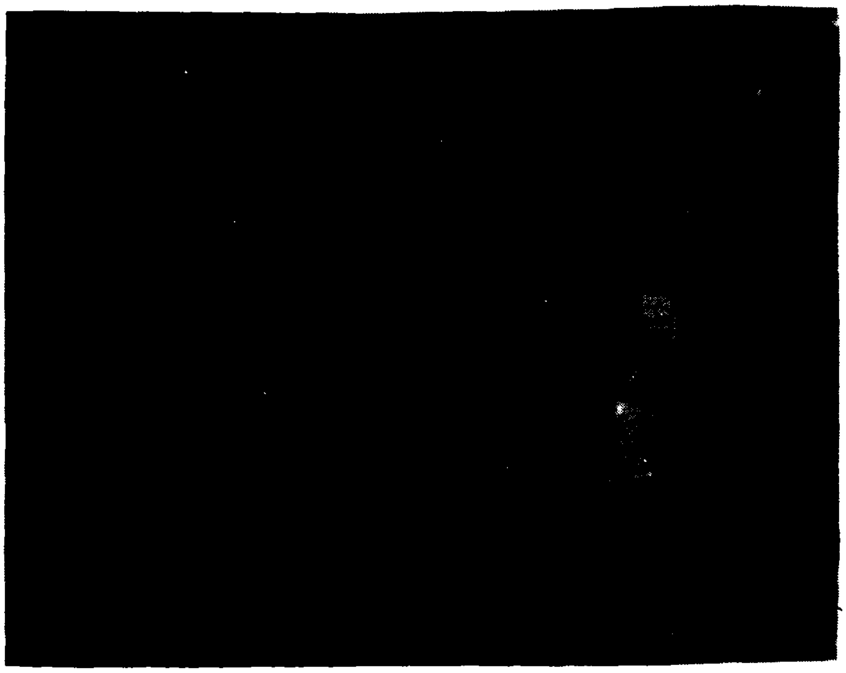
389866 CYLINDER BLOCK AFTER 713:49 HOURS OF OPERATION IN AO-8 FLUID. CYLINDER BLOCK WAS LAPPED AFTER 57:25 HOURS OF OPERATION.

(RECOMMEND UNIT LOADING AND/OR REDUCTION OF RATED SPEED FOR USE IN AO-8 FLUID)



317084 ACTUATOR PISTON AFTER
656.24 HOURS OF OPERATION IN
AO-8 FLUID

(RECOMMEND MATERIAL CHANGE
FOR USE IN AO-8 FLUID)



317105 SUB-ASSEMBLY

MOUNTING FLANGE

357850 SUB-ASSEMBLY YOKE
BALL AND PIN AFTER 713:49
HOURS OPERATION IN AO-8
FLUID

SPERRY VICKERS REPORT NO. 03-802054

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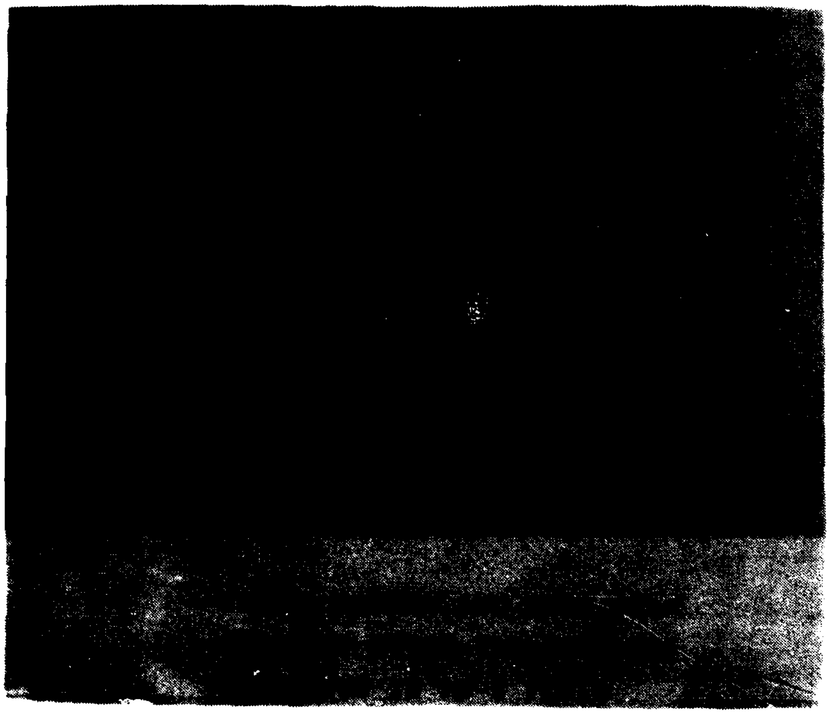

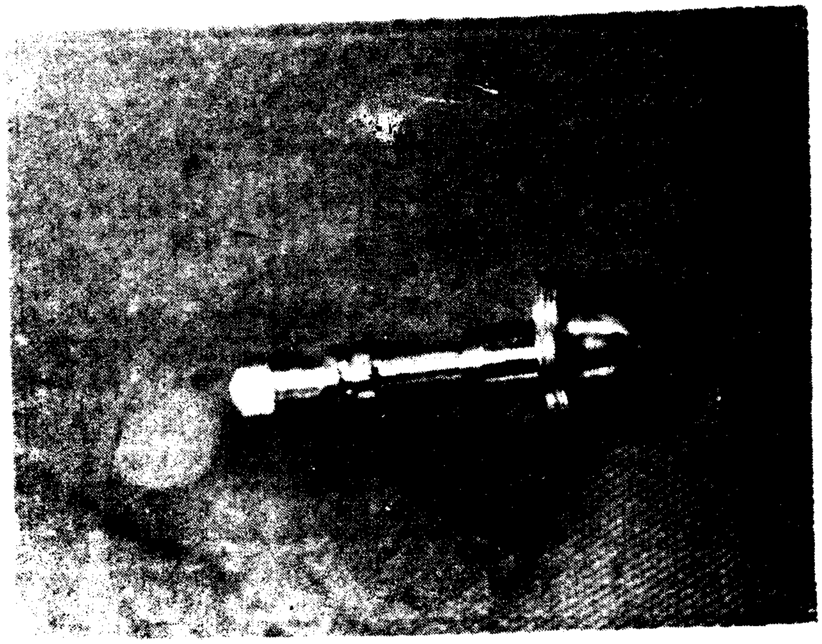

314445 PILOT VALVE AFTER
713:49 HOURS OF OPERATION
IN AO-8 FLUID

424073 SUB-ASSEMBLY VALVE PLATE AFTER 656:24 HOURS OPERATION IN AO-8 FLUID

(RECOMMEND REDUCED UNIT LOADING AND/OR REDUCTION OF RATED SPEED FOR USE IN AO-8 FLUID)

SHAFT SEAL PARTS AFTER 713:49
HOURS OF OPERATION IN AO-8 FLUID
(35123-K SUB-ASSEMBLY SHAFT SEAL)

APPENDIX J
SERVOACTUATOR TEST PLAN

The servoactuator test outlined herein will meet the requirements specified in Section F, paragraph 4.3.4 of contract F33615-76-C-2064. The servoactuator to be tested is the Borg-Warner, Nuclear Valve Division (formerly Weston Hydraulics) Part Number 27830-4 B-52 elevator power control actuator. The servoactuator is qualified to the Boeing specification part number 10-30433-503 using MIL-H-5606 hydraulic fluid. The servoactuator has been modified to provide performance and chemical compatibility to the Halocarbon Products AO-8 hydraulic fluid used in this test. The fluid seals originally of nitrile elastomer have been replaced with Firestone's "PNF" fluorinated phosphonitrilic elastomer. To allow equivalent output performance, the control valve metering slots have been widened to allow equal flow gain with the denser fluid.

All testing will be performed in a test setup as shown in Figure J1. The instrumentation will be state-of-the-art and recently calibrated. The ambient temperature will be 85 $\pm$20F and the fluid temperature will be 100 $\pm$20F, unless otherwise specified.

Test Procedure

Inspection

A teardown inspection will be performed prior to and at the completion of testing. Valve spool, rod and rod bearing wear surfaces will be visually inspected and photographed for comparison. The actuator rod and cylinder inside-diameter will be measured before and after testing for a wear measurement. The test system and servoactuator filters and all elastomer seals will be inspected at the end of testing.

Velocity Test

The servoactuator no-load velocity will be determined with both hydraulic

systems pressurized by quickly moving the input lever from stop to stop. The no-load velocity should be 6.25 inches per second (80 degrees per second surface rate).

Performance Tests

a. Dynamic Response - The servoactuator electrical-mode frequency response will be determined while mounted on the test fixture which simulates the aircraft structure compliance and control surface inertia. A sinusoidal function generator will provide input commands to the "system one" servovalve at the following frequencies while both hydraulic systems are pressurized to 3000 psi and the "system one" shutoff valve is energized open.

Test Frequency - Hz

0.1
1 to 10 at 1-Hz increments
10 to 30 at 2-Hz increments

The change in the actuator output amplitude and phase lag, and the output position versus input command signal will be determined and compared with the predicted Task I computer simulation and previously taken MIL-H-5606 test results.

The foregoing test will be repeated for "system two."

b. Hysteresis - The no-load actuator output hysteresis loop width throughout the full range of mechanical and electrical inputs will be measured. The Boeing specification requires that hysteresis not exceed .008-inch input lever displacement, nor .02-volts electrical input.

c. Threshold - The no-load actuator output threshold throughout the full range of mechanical and electrical inputs will be measured for minimum input to cause output motion. The Boeing specification requires that threshold not exceed .004-inch pilot input nor .0035-volt electrical input.

d. Leakage - All external leakage will be recorded as to amount and location. The Boeing specification limits external leakage to one drop from each piston rod seal per one hundred full stroke cycles. The internal leakage will be measured with the shutoff valves deenergized and energized. The Boeing specification requires that leakage not exceed 20 cubic inches per minute per system with the shutoff valves deenergized nor 71 cubic inches per minute per system with either shutoff valve energized.

Endurance Testing

The servoactuator will be endurance tested in a rigid fixture with the output spring-centered to simulate control surface load per the following table of conditions.

ONE UNIT OF TEST CONDITIONS*

SCHEDULE	COMMAND, in.**	FREQ., Hz	SPRING RATE, lbs/inch	CYCLES
A	± .157	1(3 optional)	50,000	400,000
B	±1.465	1	50,000	4,000
C	± .392	1(2 optional)	50,000	65,000
D	± .626	1(2 optional)	50,000	29,000
E	±1.165	1(2 optional)	50,000	5,000
F	±1.465	1(2 optional)	50,000	1,000

*The total Endurance Test consists of repeating a unit of test conditions ten times.

**Output stroke about midstroke position.

A continuous sinusoidal electrical input signal at the noted frequency to provide the stroke of Schedule A shall be applied. At 1000 cycle intervals, manual inputs to provide ten cycles of Schedule B shall be superimposed on the electrical signal. At the completion of the 400,000 cycles, manual input cycling will be applied for Schedules C, D, E, and F.

At the completion of the fifth and tenth unit of testing, the actuator will be performance tested per the procedure described above.

Low Temperature Test

The servoactuator low (minimum) temperature performance will be determined with a -65F ambient temperature to simulate actual aircraft conditions. The performance tests will consist of dynamic response, hysteresis, threshold, and leakage tests as previously described.

High Temperature Test

The servoactuator high (maximum) temperature performance will be determined per the general requirements of MIL-STD-810C, Method 501.1, Procedure I with 250F ambient and fluid temperatures. The performance tests will consist of dynamic response, hysteresis, threshold, and leakage tests as previously described. The Boeing actuator specification requires that these tests be performed with a 160F ambient and a 225F fluid temperature. Therefore, no direct comparison can be made to MIL-H-5606 data.

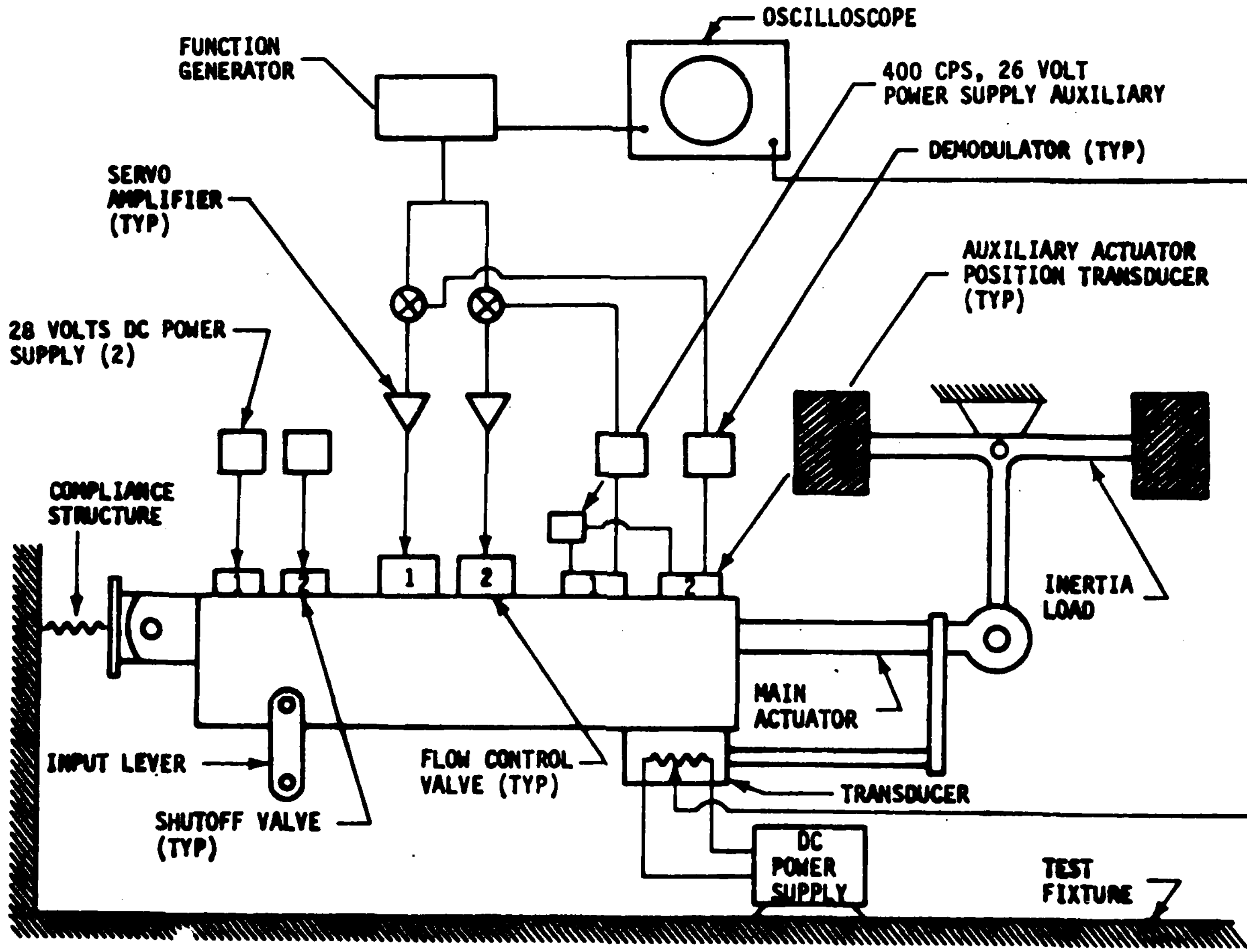

Figure J1. Servoactuator test setup

APPENDIX K

SPERRY-VICKERS REPORT NO. 03-802054
TEARDOWN INSPECTION OF PV3-075-19 PUMP
AFTER 284-HOUR SERVOACTUATOR TEST
PROJECT NO. 8-1102-218 DATED SEPTEMBER 30, 1980

1960A (3/78)

REPORT NO. 03- 802054

SPERRY VICKERS

JACKSON, MISSISSIPPI 39205

REPORT
ON

TEARDOWN INSPECTION OF PV3-075-19
AFTER 284 HOUR SERVO-ACTUATOR TEST

PROJECT NO. 8-1102-218

DATE RELEASED September 30, 1980

PREPARED BY R. A. Breckenridge
R. A. Breckenridge

CHECKED BY

APPROVED BY F. W. Perian
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CONCURRED BY

RELEASED BY H. J. Sculthorpe
H. J. Sculthorpe

APPROVED FOR OUTSIDE RELEASE H. J. Sculthorpe
H. J. Sculthorpe

ABSTRACT

The PV3-075-19 is an inline piston, variable displacement pump. It is identical to the PV3-075-15 except that the carbon shaft seal was replaced with a bronze shaft seal and the elastomer seals are PNF rather than Viton. It was run in a servo-actuator test for 284 hours at Boeing/ Wichita at a speed of 2500 RPM at temperatures from 100 to 260°F. A startup from rest to 7000 RPM was done at -65°F. Teardown inspection showed a coating of mostly carbon particles on all bronze surfaces. There was evidence of binding in the thrust bearing and in the pintle bearings.

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PAGE NO.

1.0 INTRODUCTION

A PV3-075-19 pump, Serial No. MX319687, has been run in AO-8 fire resistant fluid for a servo-actuator test at the Boeing facility at Wichita. The test was 284 hours run time, and consisted of the following conditions:

TEMPERATURE (°F)	DURATION (HRS)	SPEED (RPM)
-65	10 Min. Max.	0-7000 Start-up
100-130	247	2500
131-150	36	2500
240-260	1	2500

At the conclusion of the test the unit was returned to the Vickers facility in Jackson for teardown and inspection.

The PV3-075-19 is an inline piston, variable displacement pump. It is identical to the PV3-075-15 except that the carbon shaft seal was replaced with a bronze shaft seal, and the Viton seals were replaced with PNF.

2.0 CONCLUSIONS

All bronze parts showed a thick deposit on non-rubbing surfaces and discoloration of rubbing surfaces. Analysis of the deposit on the cylinder block showed it to be mostly carbon. Steel parts showed brown discoloration, but no thick deposit.

The indications of thrust bearing binding and possible chatter of the pintle bearings raise the possibility that the lubrication properties of the fluid are inadequate.

Steel parts run in the fluid and then exposed to air corroded rapidly.

3.0 COMMENTS ON CONDITION OF PARTS

3.1 The fluid drained from the case before disassembly contained fine, dark colored particles.

3.2 Valve Block

The valving surface, Figure 1, showed two dark brown streaks in the path of the Kingsbury pads at about top and bottom dead centers. These streaks contained outlines of the Kingsbury pads, possibly caused by the stationary contact between the valve block and the cylinder block for a period of time in the presence of the fluid.

3.0 COMMENTS ON CONDITION OF PARTS (Continued)

3.2 Valve Block (Continued)

There were three smaller spots which were initially thought to be possible smears, also in the Kingsbury path. Later, nital etch test proved that there was no softening of the material.

3.3 Cylinder Block

There was a dark brown deposit on all non-rubbing surfaces that was porous-appearing and could be scraped off with the fingernail, Figure 2. The mating surface with the valve block showed a discoloration about the same color as the deposit on the non-rubbing surfaces. It, however, could not be scraped off. Analysis of the scrapings by Mr. C. E. Snyder, Reference 1, showed them to be mostly carbon particles probably from the elastomer seals, with some chlorine, probably from the AO-8 fluid.

3.4 Yoke Actuator Piston

There was a slight polishing at the inserted end of the barrel, but no evidence of galling.

3.5 Piston/Shoe Sub-assembly

The outer diameter of the shoes had the same dark porous coating as the non-rubbing surfaces of the cylinder block. The matter also shows up in the grooves in the bottom of the shoes. The necks of the shoes and the tops of the shoe pads, where they are in contact with the hold-down plate, exhibit the same dark brown discoloration as the face of the cylinder block.

The pads of all shoes except number 6 show radial wear marks (not scratches) across the inner and outer pads, Figures 3 thru 11. Some also have small radial microscopic scratches, but there seems to be no correlation with the wear marks.

The pads of shoes 4, 5, 6, 8 and 9 have dark brown spots, suggesting that possibly the shoes are no longer flat.

The number 2 piston has a polished band around the barrel about .9 inches from the end and about .070 in width. Number 3 and number 4 have wear marks in the same place.

Number 5 has a wear mark around the piston about 1.05 from the end. Number 8 has a polished band .85 - 1.0 from the end. None of these appear abnormal.

3.6 Shoe Hold-down Plate

The shoe hold-down plate, Figure 12 has a brown discoloration lighter in color than on bronze parts. This coating can be marked with the fingernail but not removed. The areas where the plate touches the shoe flanges and the bronze plating on the retainer plate are not discolored.

3.7 Shoe Hold-down Plate Retainer

The shoe hold-down plate retainer has a dark brown discoloration on the bronze surface in contact with the hold-down plate. The discoloration also shows up on the surface in contact with the yoke, except for areas around the screw holes, where the screws held the two surfaces in tight contact.

The steel side shows a shade lighter discoloration then the other steel parts.

3.8 Yoke

The yoke had a brown discoloration under the high-pressure side of the wear plate.

3.9 Shoe Wear Plate

The wear plate had superficial shoe outlines all around the part, Figure 13, of the same nature as the Kingsbury pad outlines on the valve block. Portions of the top surface not worn by the shoes had the same brown discoloration as other steel parts. There was very little discoloration on the other (yoke) side, except on the high-pressure side. This pattern matched that on the yoke.

3.10 Drive Shaft Bearings

The drive shaft thrust bearing was slightly rough when removed, but got much rougher when dry. There was a dark brown band around the outside of the outer race, Figure 14, and a brown spot in the bearing bore at about the bottom dead center position, as if the race were spinning. Measurement of the balls in this unit after disassembly showed diameters from .3434 to .3439, with some out of round by as much as .0004. Some of this variation could have come about as a result of corrosion after contact with air, but if balls were out of round in the unit, binding could result, explaining the apparent spinning of the race.

The needle bearing race was in good condition.

3.11 Compensator

The compensator was in good shape. The spindle of the spool had a light brown discoloration.

3.12 Shaft Seal

The bronze shaft seal had a thick brown coating similar to the cylinder block. This coating was also on the chrome plating of the seal, except where the grommet was in contact.

3.13 Pintle Bearings

The pintle bearings were slightly rough. Both the inner and outer races had brown marks perpendicular to the bearing bath. They were not scratches, but could possibly be chatter marks. They were too close together to be impressions of the rollers.

SPERRY VICKERS

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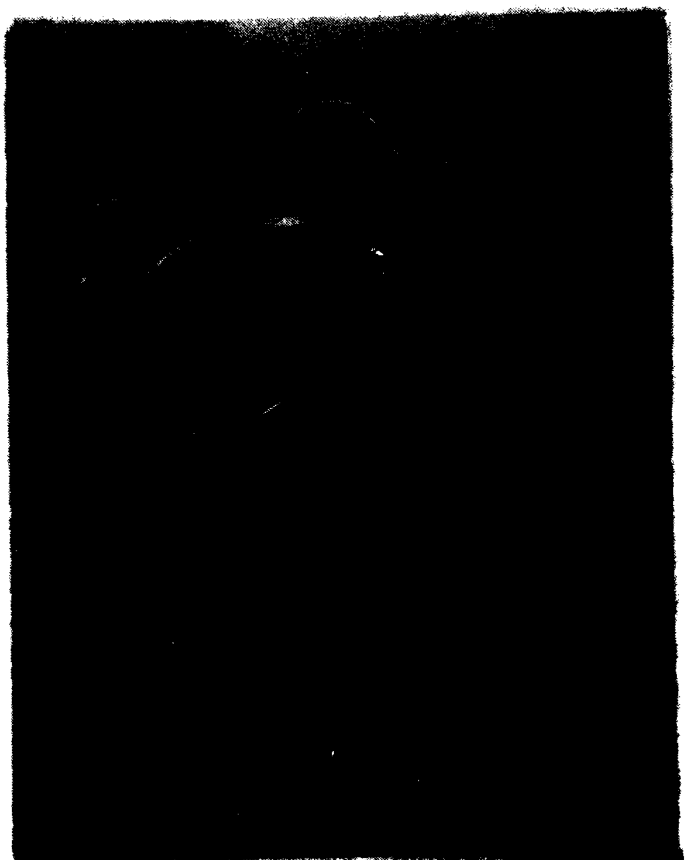

FIGURE 1 - VALVE BLOCK

PART NUMBER [illegible]

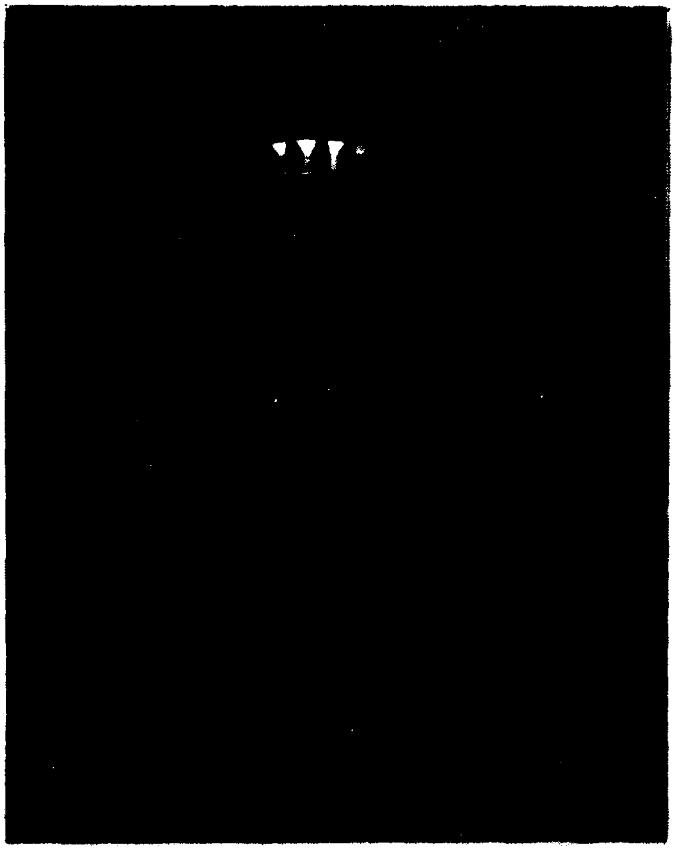

FIGURE 2 - CYLINDER BLOCK

PART NUMBER 389866

SPERRY VICKERS

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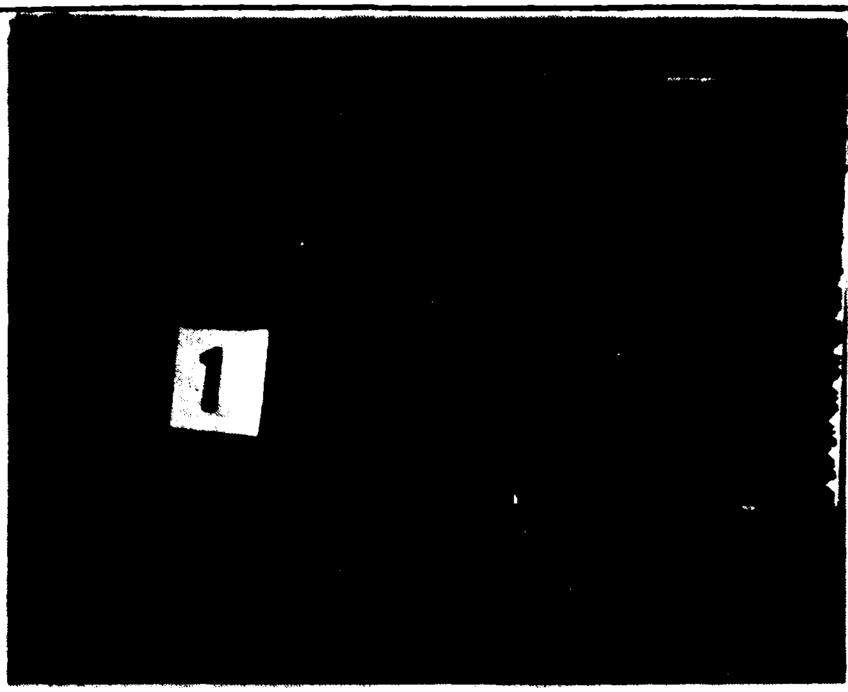
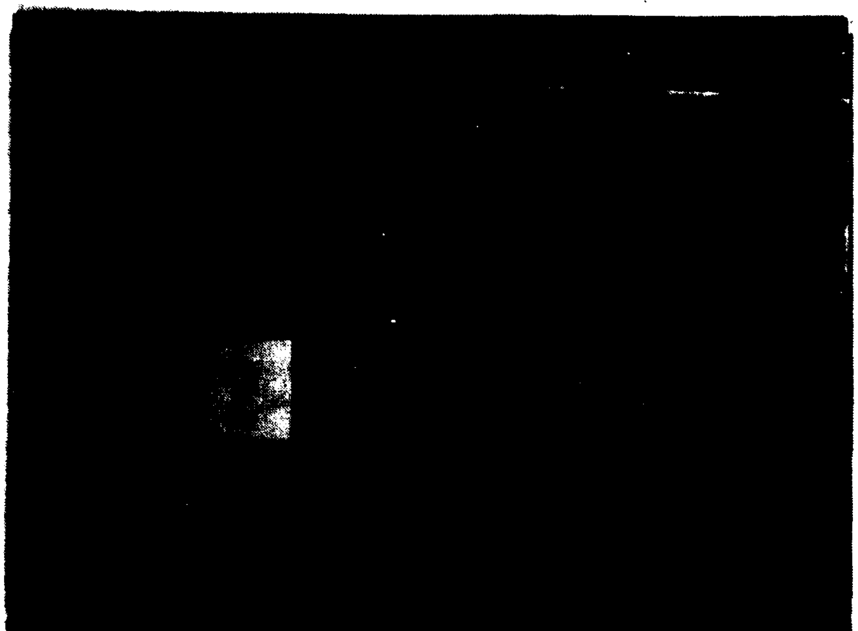


FIGURE 3 - PISTON SHOE NO. 1
PART NUMBER 438200



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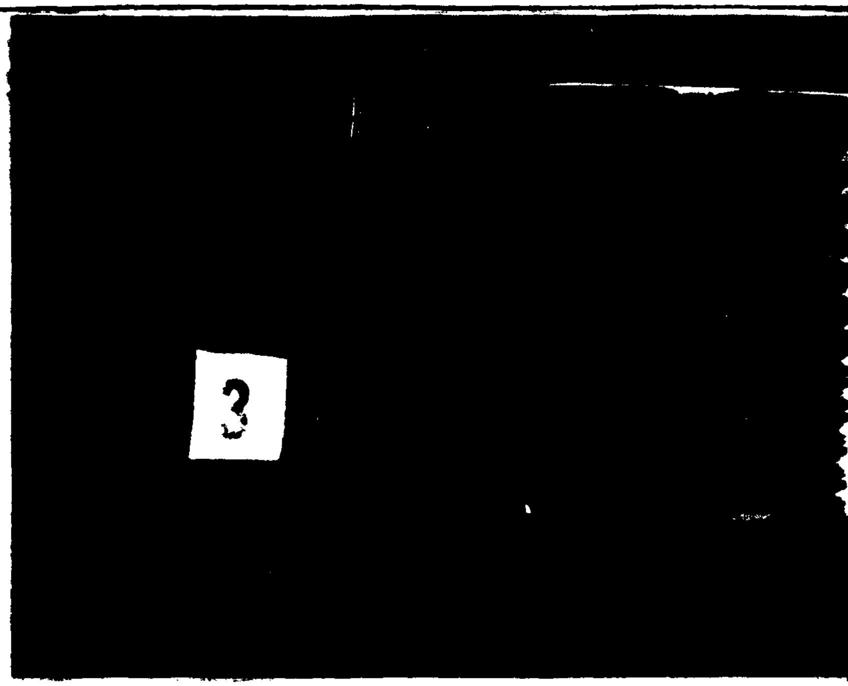


FIGURE 5 - PISTON SHOE NO. 3

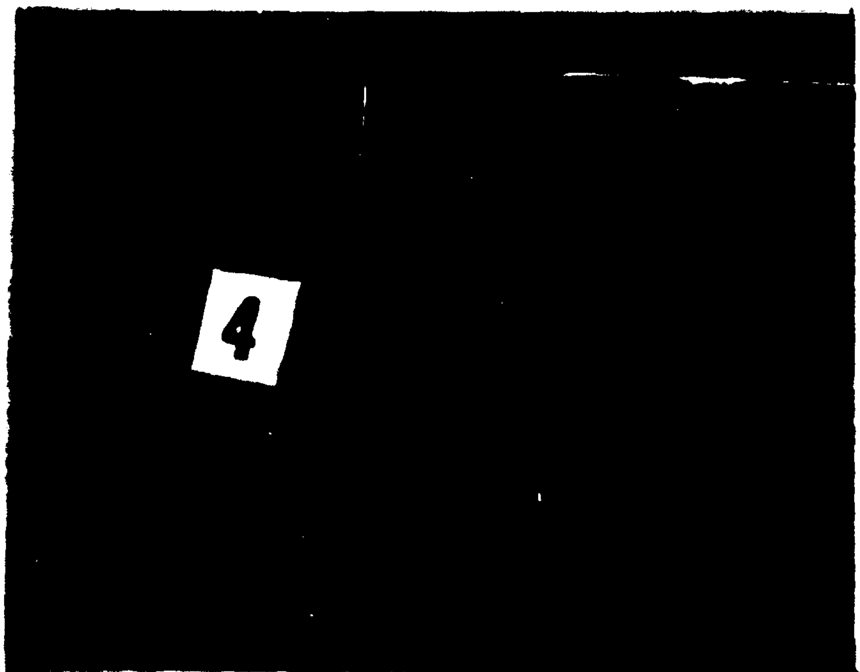


FIGURE 6 - PISTON SHOE NO. 4

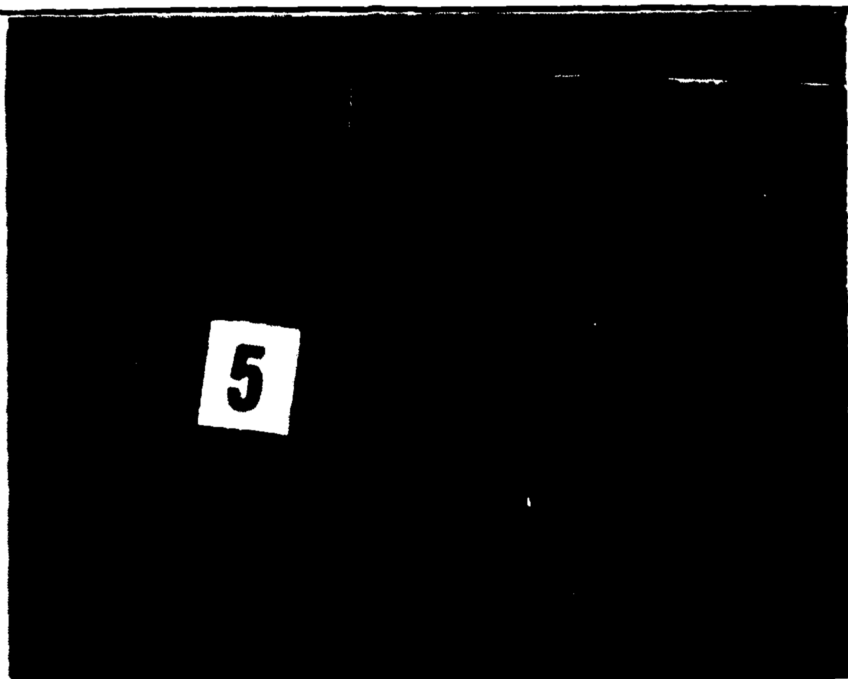
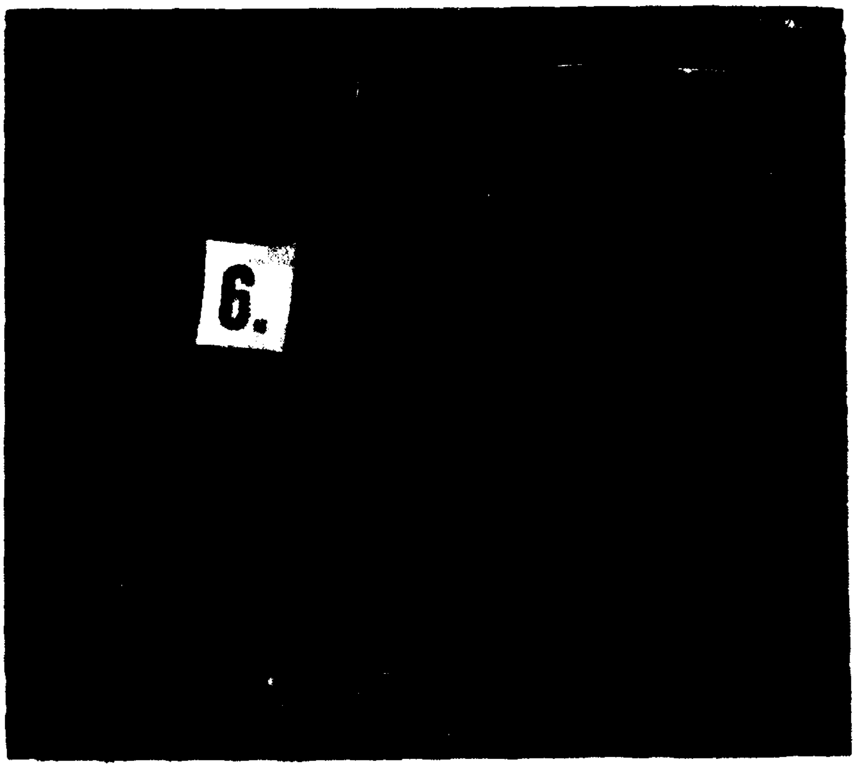


FIGURE 7 - PISTON SHOE NO. 5



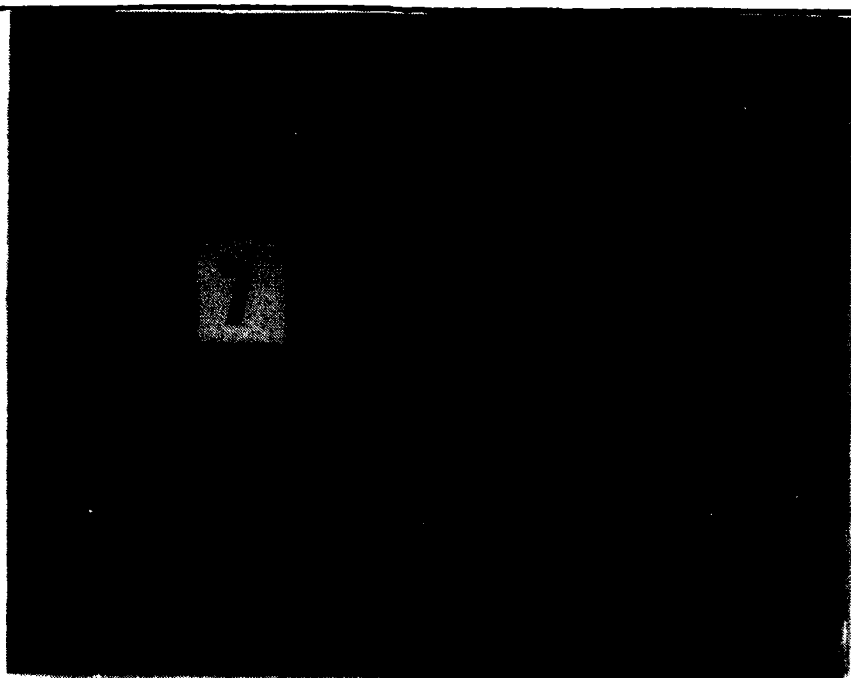


FIGURE 9 - PISTON SHOE NO. 7

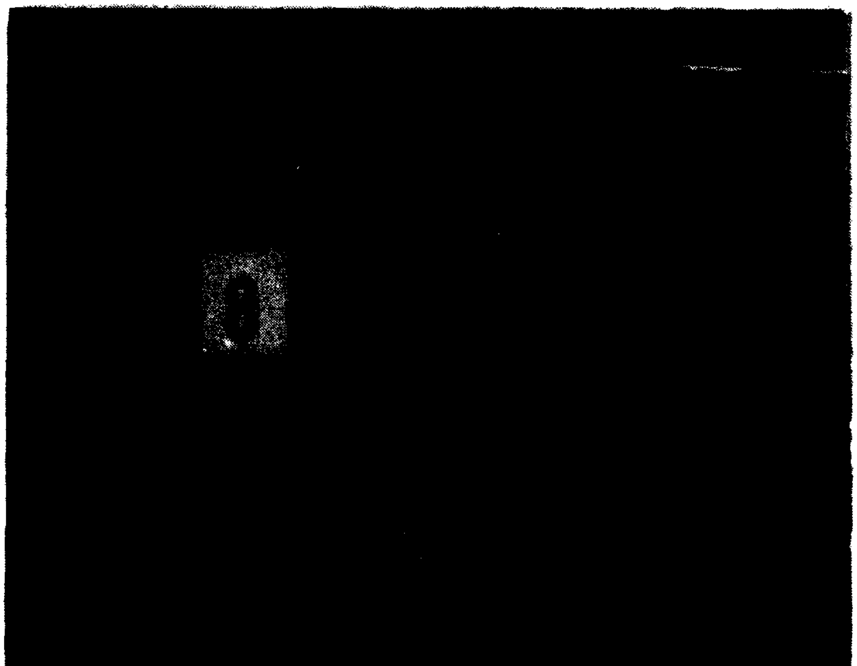
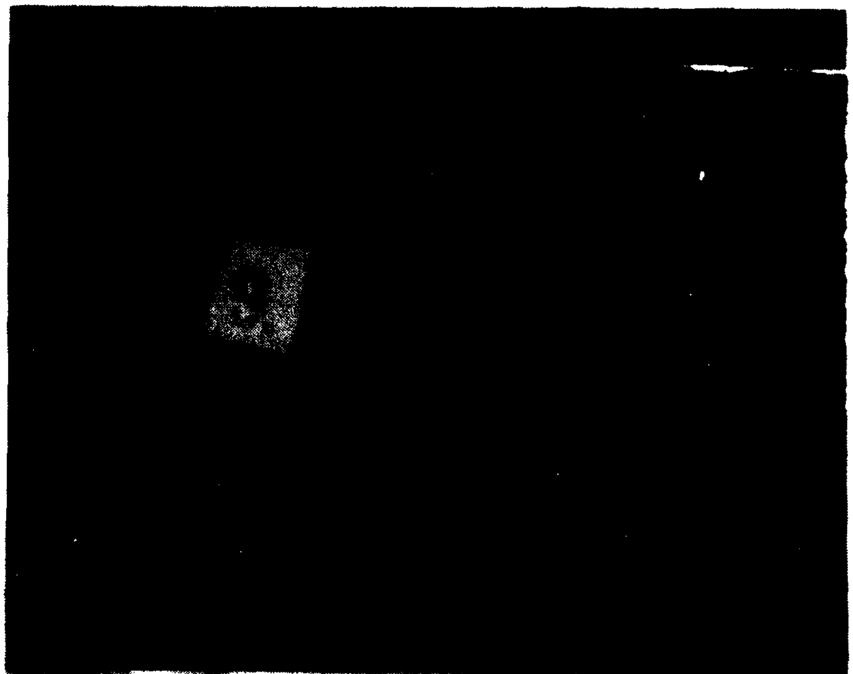
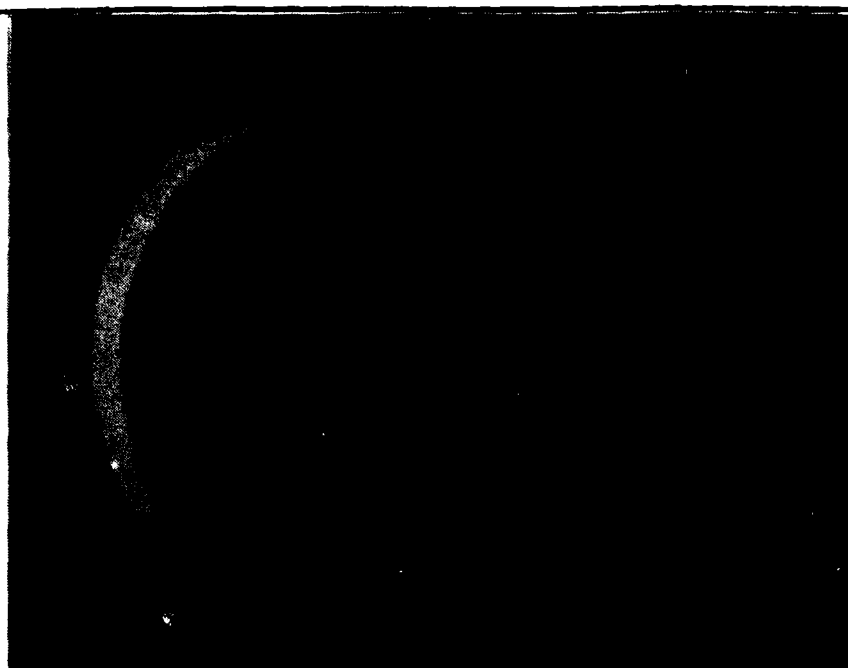
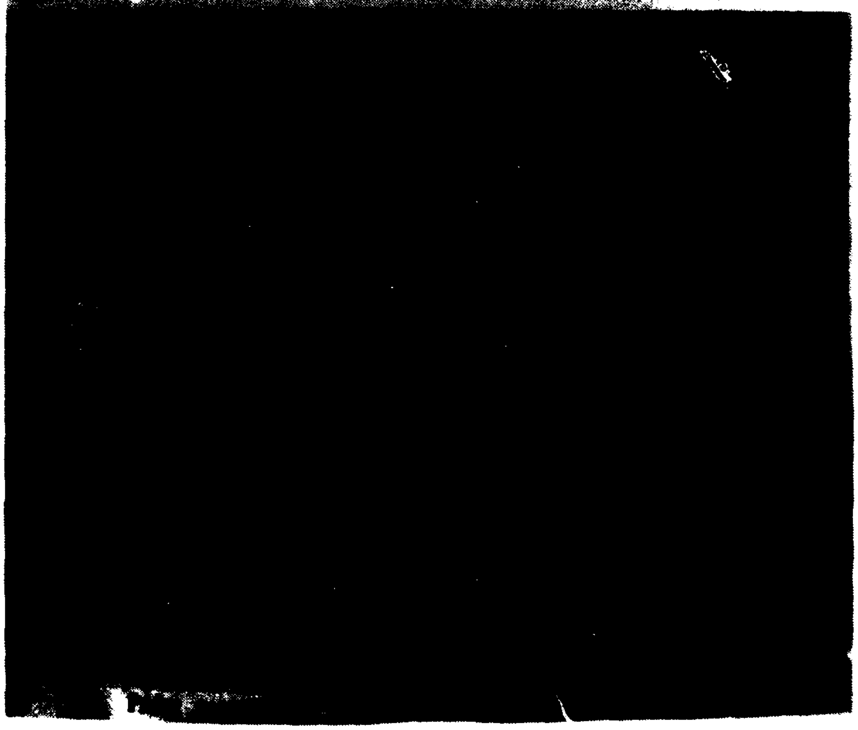


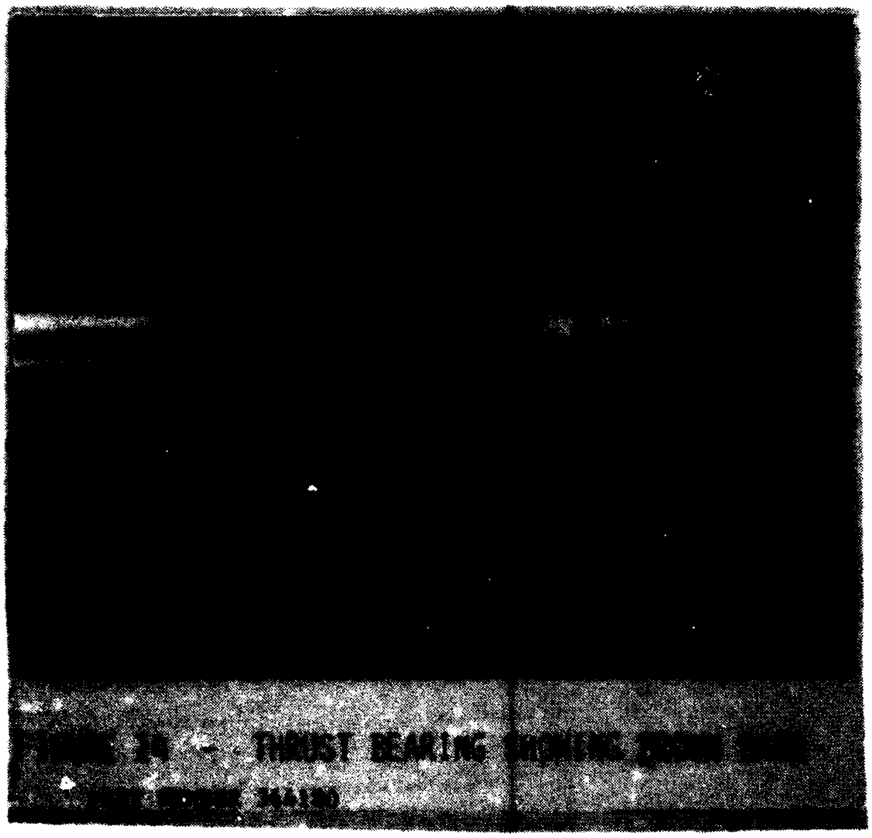
FIGURE 10 - [illegible]



[illegible] - PISTON [illegible]






10 - THRUST BEARING

REFERENCES

1 Letter to B. Campbell from C. E. Snyder, "Characterization of Black Particles from Boeing Long Term Servoactuator Test; dated 11 June, 1980.

Distribution List

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APPENDIX L

PALL CORPORATION TECHNICAL REPORT NO. 200
COMPATIBILITY STUDY BETWEEN APM FILTER MATERIALS
AND A CANDIDATE FIRE RESISTANT HYDRAULIC FLUID
FOR THE U.S. AIR FORCE
DATED DECEMBER 21, 1978

Technical Report No. 200

Prepared for

Bruce Campbell

Company

Wright-Patterson Air Force Base

Date

December 21, 1978

AD-A118 169 BOEING MILITARY AIRPLANE CO SEATTLE WA F/G 11/8
FIRE RESISTANT AIRCRAFT HYDRAULIC SYSTEM, (U)
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SCIENTIFIC & LABORATORY SERVICES DEPT.
Director: Erwin Kirnbauer
FILTER PERFORMANCE TESTS
PROCESS & SYSTEM CONTAMINATION CONTROL
PARTICULATE & MICROBIAL FLUID CONTAMINATION ANALYSIS
WEAR ANALYSIS
BIOLOGICAL CONTROLS

December 21, 1978

Report #200

Compatibility Study Between APM Filter Materials and a Candidate Fire Resistant Hydraulic Fluid for the U.S. Air Force

1. OBJECTIVE

To determine the compatibility between materials used to construct APM hydraulic filter elements and an Air Force candidate fluid for aerospace fire resistant hydraulic fluid applications.

2. MATERIALS

2.1 Samples of media representative of material used in constructing APM hydraulic elements.

2.2 Complete filter elements P/N AC900F-6UP1. The materials used to construct this filter elements, including side and end cap seals, are typical of APM hydraulic filters.

2.3 Candidate Air Force hydraulic fluid: Halocarbon A08, batch 33178, manufactured by Halocarbon Products Company.

2.4 Mil-H-5606C, used as a reference fluid.

3. EXPERIMENTAL PROCEDURES

3.1 Duplicate samples of filter media were heat soaked in Halocarbon A08 for 72 hours at 275°F. Similarly, filter media samples were heat soaked in Mil-H-5606C.

International Offices & Plants: Pall Europe Ltd., Portsmouth; Pall GmbH, Frankfurt; Pall France SA, Paris; Pall Italia S.R.L., Milan
Pall (Canada) Ltd., Ontario; Pall Far East Corp., Tokyo; Marusho Ltd., Tokyo.

3.2 One filter element was similarly heat soaked in each of the hydraulic fluids.

3.3 After heat soaking the media samples were then cleaned and oven dried. Changes in weight before and after heat soak were measured using a Christian Becker Style AB-4 Analytical Balance.

3.4 Measurements of tensile strengths of the media samples after heat soak were made using an Instron Model 1130 Tensile Strength Test Apparatus.

3.5 Filter elements were visually examined for deterioration after heat soak.

4. RESULTS

4.1 Filter Media Samples

Test Fluid	Weight Change
Halocarbon A08	Negligible
Mil-H-5606C	Negligible

Test Fluid	Tensile Strength Change
Halocarbon A08	No Change Detected
Mil-H-5606C	No Change Detected

4.2 Visual examination of heat soaked filter elements showed no deterioration.

5. CONCLUSIONS

The compatibility study of APM filter material with the Air Force Candidate fluid indicated Halocarbon A08 is an acceptable fluid which does not cause deterioration of materials used to construct APM filter elements. The compatibility of the Halocarbon A08 fluid with these filter materials compare favorably with the compatibility of Mil-H-5606C.

W.N.N.

William N. Needelman
Staff Scientist

U.S. GOVERNMENT PRINTING OFFICE: 1982-559-064/3066

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